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A PROCEDURE AND CASE STUDY DEMONSTRATIONS FOR EVALUATING THE COST OF THERMAL EFFLUENT CONTROL FOR PROPOSED STEAM-ELECTRIC GENERATING UNITS

Frank Austin Smith, et al

Center for the Environment and Man, Incorporated

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**A PROCEDURE AND CASE STUDY DEMONSTRATIONS
FOR EVALUATING THE COST OF THERMAL EFFLUENT CONTROL
FOR PROPOSED STEAM-ELECTRIC GENERATING UNITS**

Final Report

for the

National Thermal Pollution Research Program

EPA, WQO Contract No. 14-12-087

**An Economic Analysis of Thermal Pollution Abatement Costs
in the Electric Power Industry**

**CEM Report No. 0000-405
May 1972**

**Frank Austin Smith,
Principal Investigator
Leonard Ortolano,
Consultant**

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in the Electric Power Industry

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by
Frank Austin Smith
and
Leonard Ortolano

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THE CENTER FOR THE ENVIRONMENT AND MAN, INC.
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FOREWORD

The research and development project reported on herein was undertaken for the National Thermal Pollution Research Program in response to a perceived need for improved information and methods relevant to the estimation of economic costs of thermal control at steam-electric generating stations. The stated objectives of the project were threefold:

- To develop a conceptual approach and an operational procedure for evaluating a broad array of technical alternatives for thermal effluent control at the individual plant level. The procedure should be capable of determining least-cost options for achieving alternative thermal effluent standards, consistent with regional environmental, factor input cost, and utility-system design considerations.
- To survey and develop current empirical information on cooling system component costs, meteorological design data, and other factors relevant to the design and costing of cooling system and heat rejector components.
- To demonstrate the capabilities of our cost-estimating procedure and produce explicit cost estimates for a limited number of realistic U.S. case study situations.

The scope of the project has been explicitly and realistically limited in a number of respects. Among the more important of these self-imposed restrictions have been the following:

- No attempt has been made to undertake a general optimization study for steam-electric plants encompassing all aspects of site-selection, plant sizing, load planning, system interconnection issues, and the like. Nor have we explicitly considered other environmental issues, such as air pollution control, transmission or radiation and other plant safety considerations. Thus, our study should properly be regarded as one pertaining only to the "sub-optimization" aspect of thermal effluent control. The other features of the broader electric utility and environmental systems, within both of which the generating station resides as a component element, thus enter our perview only, if at all, as parametric design constraints that may be varied for purposes of sensitivity analysis.
- No formal attempt has been made to deal with the technically related questions of effluent temperature dispersion in the aquatic (or atmospheric) receiving environment or the ecological and social cost implications of thermal discharges

per se. However, we have structured our approach so that it could be integrated with models of aquatic temperature dispersion, and also so that it is consistent with various models for social control such as effluent standards or effluent charges schemes.

- Potential possibilities for beneficial utilization of "waste" heat in either off-site or in in-situ environmental applications were taken to be outside the scope of our project, although they are not necessarily incapable of future incorporation.

Although not originally excluded by intent, we were not able during the course of the project to integrate a satisfactory method for dealing with the complex questions of seasonal variations in atmospheric, hydrological and social control variables. Thus, it should be clearly understood that the design and cost-estimation procedures and results described in the body of our report rely more heavily than they probably should on the conceptual basis of annual design conditions. As we argue in Section I of the report, we do not feel that this shortcoming in our scope seriously compromises the results of our efforts in the present context.

During the early planning phase of the project it was recognized that the differences between existing plants (i.e., those in the later stages of construction or already on line) and proposed (or scheduled) units in the earlier phases of planning and design were sufficiently great to require separate analyses. In particular, numerous technically feasible abatement options (such as changes in design capacity, thermal efficiency, or condenser redesign) are generally to be ruled out, a priori, in the case of existing plants. More importantly, issues relating to system interdependencies and problems of estimating operating relationships under off-design conditions cause rather significant differences from an engineering-analysis standpoint. Our work on existing plants has not been carried as far as that for proposed units, partly due to our sense of priorities and partly due to the above related technical problems that were not fully appreciated during the early phases of our work.

We have thus chosen not to include the preliminary results of our work on existing plants in the present report, which consequently deals exclusively with proposed generating units. It seems likely that few, if any, new units scheduled to come on line during the remainder of the present decade will be entirely exempted from effluent control considerations; and many if not most will face quite restrictive

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public policy conditions. On the other hand, it seems much less likely that many existing units (particularly the older and smaller ones) will face retrofitting or other control option requirements. Thus, we feel that analysis for proposed plants relates to the more important and relevant of the two thermal abatement situations.

The body of this report is contained in three sections. Section I provides an introductory overview of our general approach and methods utilized in developing and applying the cost estimating procedure. Section II presents a demonstration of the procedure together with results from its application to our empirical case study situation involving a prototype 547 mw coal-fired generating unit. Section III provides a similar, although less extensive, case study demonstration for a 1000 mw prototype nuclear unit, as described.

Readers interested in pursuing in greater depth the technical engineering-economic design characteristics and assumptions underlying our computerized procedures for designing and evaluating alternative cooling system options are referred to Appendix A, which provides detailed technical specifications. Appendix B describes the approach utilized to develop mathematical cooling tower cost-estimating functions based on available published data. Appendices C and D provide additional detail on the particular design input values utilized in producing the cooling system design analyses for the two case study units demonstrated in Sections II and III.

CONCLUSIONS

This study has developed a conceptual approach and an operational procedure for evaluating alternative thermal effluent modification programs and estimating the generating station incremental costs of achieving alternative effluent standards. The procedure is versatile in that (1) it includes a broad range of potential technically feasible alternatives for design modifications both to the steam-cycle and the cooling system components of the generating unit, and that (2) it relates simultaneously to both effluent temperature and effluent flowrate measures of plant abatement performance variables.

The most important conclusions pertaining to the procedure itself are that:

- It has been demonstrated by application to two case study plants and shown to be feasible and operational, and,
- in its present form the procedure has produced design results that are generally consistent with those observed in practice, and cost results that appear consistent with hypothesized functional relationships and of the correct general orders of magnitude consistent with various cost input parameters.

For the above reasons we conclude that, in spite of certain present shortcomings explicitly described in the case study discussions, the procedure has worthwhile potentials both for research purposes and as a tool in water quality management applications. Additional work on verification and computer program refinement appears worthy of consideration. The procedure in its present automated form does not include seasonal variation analysis nor does it produce present values on a discounted basis. The former would involve a considerable effort, but the latter would seem to be a relatively routine modification.

The following conclusions regarding abatement costs emerge from the case studies:

- As expected on the basis of economic theory, thermal effluent modification costs increase at an increasing rate as functions of both decreases in design effluent temperature and decreases in design effluent flowrate with respect to base values of these effluent characteristics generally encountered in the northeastern United States.

- The order of magnitude of annual abatement costs for the most restrictive effluent standards is substantial in absolute terms. Total annual cost values in our case studies for these low temperature, low flowrate standards ranged from about \$300,000/year for a 550 mw coal-fired plant to over \$2 million/year for a 1000 mw plant. (The nuclear case, however, was developed on the basis of high-cost input assumptions while the fossil-fueled case was analyzed on the basis of lower costing parameters. Thus, no comparative significance should be based on these particular values.)
- Significant partial degrees of thermal effluent control short of the most restrictive controls can often be achieved at relatively nominal cost involving only a small fraction of the cost of extreme control. This suggests that much more attention should be paid to the question of partial as opposed to closed-cycle alternatives.
- Efficient abatement programs for proposed plants of a given generating capacity will generally require modifications in both steam cycle and cooling system components of a base case plant designed in the absence of public effluent control regulations. The most important design variable in this regard appears to be the condenser vacuum (turbine back pressure). Abatement costs may be more than doubled by selection of non-optimal design vacuums.
- In all instances absolute values of abatement costs are sensitive to input costing factors and many other design conditions. It, therefore, seems desirable to develop further the cost ranges implicit in alternative combinations of input values via additional sensitivity analyses for these and other case study plants.

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While no claim can be made for precise optimization for our procedure, it is felt that it does yield reasonable first approximations to the designs that would actually result from detailed, final-optimization engineering design projects. More important from a research standpoint, it yields considerable insight on questions of how various cost and performance variables can be expected to shift with variations in effluent quality standards.

ACKNOWLEDGMENTS

Credit should be acknowledged to a number of individuals who made important contributions during the course of the project, both to our conceptual and technical understanding and to the substance of the results contained herein. Mr. H.L.R. Newbold served as our principal engineering design consultant under a subcontract to Jackson and Moreland Division of United Engineers and Constructors, Inc. He was primarily responsible for carrying out the engineering design analysis and capital cost calculations pertaining to the steam-cycle design modifications performed in connection with both case-study plants. Perhaps even more importantly, he provided a continual sounding board for our ideas and a crucial source of practical experience and professional expertise in matters of generating station design practice that would otherwise have been lacking to the project. Many important criticisms and valuable suggestions were also provided during the developmental stages of our work by two other project consultants, Dr. Blair Bower of Resources for the Future, Inc., and Dr. George Löff of Colorado State University. Both of these men brought a considerable range of engineering experience and background in the field of water resources management to bear on many of the conceptual and practical design aspects of our methodology.

A number of individuals directly associated with the electric utility industry, either from an operating utility company, design engineering or specialized equipment vendor perspective, also contributed valuable insights or specific design information. Among those to whom we are most indebted are Mr. Robert Evans of Northeast Utilities, Mr. Arthur Shepard of Vt. Yankee Nuclear Power Co., and Mr. R.E. Kates of the Marley Co.

Miss Marilyn Sorenson, formerly of the CEM staff and currently at the University of Connecticut, carried the main burden of the computer programming effort required for the project, and we are grateful for her conscientious and skilled work.

Among the many individuals associated with the Environmental Protection Agency who have provided assistance and critical review, special note should be made of the contributions of Mr. Frank H. Rainwater, Chief of the National Thermal Pollution Research Program, who served as Technical Project Officer for EPA. Mr. Robert Brown, formerly of the Systems Analysis and Economics Division and presently in the Water Systems Division, represented the program interests of the latter EPA offices, along with Mr. Edwin Johnson who served as our initial Technical Monitor in the Systems Analysis and Economics Division.

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1. INTRODUCTION: APPROACH AND METHODS

The present project has been designed to explore thermal pollution abatement possibilities for steam-electric generating plants and to develop methods for deriving quantitative estimates of associated abatement costs. This report presents the results of our work pertaining to "proposed" (or planned) generating units. In this introductory section we first discuss some of the more important operating definitions and assumptions underlying our general cost-estimating approach and then we provide a brief overview of the procedures developed to produce specific design and cost evaluations.

Operating Definitions and Assumptions

We require at the outset an operationally useful definition of thermal pollution and its abatement (reduction) to serve as a basis for assessing individual plant performance results. Ideally, such a definition should be:

- readily capable of quantitative expression
- directly relatable to any steam-electric station's basic waste heat production and disposal activity
- relevant to the immediate concerns of water quality management agencies
- as simple as possible, to facilitate comparative analysis and, at the same time, relate the essence of the problem in sufficient detail to be of general practical usefulness.

On the basis of these criteria, we have chosen to focus on the generating station's cooling water effluent as the most useful point for defining and measuring a plant's pollution performance for purposes of this study. Our working definition also involves a further abstraction from the complexities of real-world thermal pollution situations in that we consider only two of the many possible characteristics of a thermal effluent, namely flowrate (L_e) and temperature (T_e) as our physical measures of performance.

In adopting this approach, we are fully aware that many other aspects of a plant's circulating water flow and effluent discharge configuration may often have a significant bearing on consequent ecological effects [1; 3]. We are thus cognizant of the fact that our two-dimensional measure of performance may not, in itself, provide a direct indication of either ecological or social damage cost implications of a power plant's

operations. However, we are prepared to argue that these represent the two most important aspects of the power plant's relationship to the aquatic environment. Perhaps equally important, both of these variables are capable of extensive modifications, both individually and simultaneously, in response to steam-electric plant design changes. This enables us to deal quite readily with variable abatement levels. Most work in the field of thermal abatement cost analysis has dealt exclusively with questions of virtually complete control via closed systems involving cooling towers or ponds. We know of no work to date in this field that has attempted to deal either with variable or partial abatement performance levels or with specific reference to both flowrate and temperature-rise performance aspects of residual heat disposal.

In this study we thus define thermal pollution abatement in terms of changes in a plant's "desired" effluent flowrate and temperature in order to meet explicitly imposed constraints on these variables. Given proposed values of L_e and T_e , determined in the absence of effluent constraints, the degree of abatement is thus defined in terms of differences, $L_e - L^*$, and/or $T_e - T^*$, where L^* and T^* represent the maximum allowable value for effluent flowrate and temperature. The cost of abatement is thus defined as the increased cost of electricity generation attributable to the constraint(s). If we posit alternative potential values of L^* and T^* , and if we can evaluate the costs of meeting these alternative abatement levels, we can generate functional relationships between abatement costs and degrees of effluent control. This, in fact, is the principal objective of our cost estimating procedure.

For existing steam-electric plants, the levels of L_e and T_e existing in the absence of thermal discharge regulations are observable and readily measurable. In the case of proposed plants, however, determination of the "true" base levels for purposes of measuring potential abatement and abatement costs may be difficult to determine in the absence of an engineering optimization study to determine what the most economic design for the plant would be in the absence of effluent constraints. This conceptual difficulty is touched on further below. In any event, the determination of base levels is an important prerequisite to the definition and measurement of both physical abatement performance and the economic cost of abatement.

It should be observed that our conceptual approach is most directly relevant to an effluent standards viewpoint of social control. However, when coupled with physical

diffusion models it can be regarded as the first stage model in a methodology for evaluating the costs of alternative, in situ, water quality temperature standards. Alternatively, it is amenable to potential applications in evaluating a plant's abatement response to a Kneese-type effluent charges system of social control [4].

An Overview of the Cost Estimating Procedure

Basically, our procedure is designed to produce approximate technical design specifications and cost estimates for alternative programs of thermal effluent control that might be imposed on prospective steam-electric generating units. In other words, given a proposed installation designed in the absence of constraints on its cooling water effluent, with proposed effluent flowrate, L_e , and effluent temperature, T_e , we seek to determine the least-costly design modifications for satisfying any potential effluent constraints of the form $L^* < L_e$ and/or $T^* < T_e$. The additional cost of electricity generation required to meet such a constraint serves as our operational definition of the "cost of thermal effluent control" or, alternatively, the "cost of thermal pollution abatement."

In principle, our cost estimating procedure is relatively straightforward, involving five basic steps:

- (1) Determine the design characteristics of the base-case proposed unit and its environmental setting;
- (2) Evaluate alternative design modifications to the steam supply and turbo-generator (steam cycle) components of the unit and calculate the incremental costs of these alternatives;
- (3) For each of these individual steam cycle alternatives (including the base design) evaluate a series of technically compatible cooling system design options for achieving alternative cooling water effluent characteristics (involving variations in both L_e and T_e), and determine the incremental cost of each such cooling system design relative to the cost of base-case cooling system.
- (4) Specify one or more alternative "effluent standards" (combinations of L^* and T^*), and determine which among the set of cooling system design options for each potential steam-cycle option is capable of meeting each such effluent constraint at the least incremental cooling cost. (In effect, this sub-optimization step reduces the field of relevant cooling system options appropriate to any given steam cycle to those cooling systems determined to be most cost-effective in meeting each alternative effluent standard.)

- (5) Integrate the incremental cost results of Step 4 with those of Step 2 to select, from among all steam cycles and respective cost-effective cooling system combinations, the least costly combined system for achieving each alternative effluent standard. This synthesis across both major sub-systems provides for a higher order approximation to the optimal total generating system design for achieving any desired effluent standard.

Depending on desired levels of sophistication and detail, carrying out the general procedure could become extremely complex in practice, owing to the large numbers of possible technical design alternatives and variables that could be entered into the analysis, and also due to the problem of maintaining technical consistency and compatibility among various interrelated components of the generating system. However, as will become evident in the following discussion, the principal difficulties from an engineering economic standpoint arise in connection with Steps 2 and 3 (Steps 4 and 5 involving only routine computations); and the major effort of our study has been in developing a computerized methodology for evaluating large numbers of potential cooling system design variations in a relatively detailed and efficient manner.

Determining the Base Case Design

There are a number of possible approaches to selecting the base case generating unit for research or demonstration purposes. One of the most frequently employed in the engineering literature is to postulate an idealized or "typical" unit. The design characteristics for such a hypothetical case could be developed either very crudely on the basis of rough approximations by reference to "current practice," or at a very high level of detail by carrying out a full-fledged (and very expensive) engineering-optimization design analysis of the type performed in designing real systems.

A second approach, and the one found to be most expedient for use in the present study, is to take the base case technical design and cost characteristics from an actual real-world prototype unit. This has a particular advantage in that one can generally acquire the necessary information at almost any desired level of detail without engaging in detailed optimization studies; and one can reasonably assume that the design in question represents a technologically consistent and economic choice for its contemporary cost and locational factors. However, since we are concerned with proposed plants, it seems important for general research purposes to focus on prototype units that are either in the planning or early construction stages in order to

maintain a reasonable currency in terms of both technology and cost conditions. Potential users of our procedure are often likely to be those concerned with particular real plant proposals and construction site situations, and for them, the base case may thus be predetermined.

It seems evident that most new units coming on line or in the planning stages in the 1970's will in fact have been designed in the presence of some degree of thermal discharge regulation and will thus not satisfy our "pure" definition of a "privately optimal" plant for purposes of serving as a base for calculating the incremental costs attributable to publicly imposed thermal regulation per se. Here we have two options. One is to concern ourselves only with calculating the additional costs attributable to alternative higher levels of effluent control. The other is to attempt to simulate what the design characteristics and costs would have been in the absence of any thermal quality standards—i.e., to "redesign" the prototype in order to remove the built-in influences of current effluent controls.

If nothing else, this discussion should serve to emphasize the inherent problems of defining and measuring pollution abatement and abatement costs for plants not yet operating and especially those designed in advance to meet effluent discharge regulations. The two case study units selected to illustrate our procedure were, as far as we could determine, not designed to meet specific effluent or in-situ stream standards. In any event, for present purposes we have assumed that they represent "pure" examples of "privately optimal" designs against which alternative technical modifications and associated costs can be compared and measured.

Generating Unit Design Modifications

Given a base-case plant, we hypothesize alternative design modifications that might be expected to have an influence on its thermal effluent performance. These might include, for example, attempts to increase plant thermal efficiency in order to reduce its residual heat discharge rate per unit of capacity, or a reduction in design capacity itself. Another alternative might be to alter its annual operating schedule in order to generate a lower summer heat discharge.

In addition to design modifications to the steam cycle (boiler and turbo-generator) there are also, of course, numerous possible ways of altering cooling system characteristics in order to change design values of effluent flowrate and temperature. These are

developed separately in our procedure, and we discuss them separately below. These two components of the overall system cannot be considered in isolation, since they are highly interdependent. However, since the steam cycle and cooling water system are linked at the condenser and can thus have the same condenser design conditions in common—i.e., design steam condensing temperature and design value for total heat transfer—we have found it possible to separate the two components for purposes of certain design analyses while holding these two mutual design conditions as given. Our principal reason for adopting this approach was to enable us to computerize the cooling system evaluation procedure. While ideally it might have been desirable to computerize the entire procedure, including steam cycle components, we considered this impractical within the project scope and budget; and we were further discouraged from attempting this by numerous engineering design specialists who felt that it would not be feasible.

Thus, a given steam cycle design developed manually against a given design condenser vacuum (and other assumptions), and the same design condenser vacuum together with the condenser duty developed from the steam cycle analysis become the principal condenser design conditions as a basis for evaluating a series of cooling systems. All cooling systems designed on the basis of these given values are thus technically compatible with this particular steam cycle option.

For each of our case study analyses described in Sections II and III below, our engineering consultant developed a series of steam cycle options involving variations in design condenser vacuums. For the fossil-fuel case presented in Section II, he also produced variations in design thermal efficiency while holding vacuum given. While the specifics of the steam cycle modification analysis are better left until later, it is important to emphasize that this constitutes an inherently important aspect of our overall procedure, and that minimum total cost abatement options cannot generally be developed for proposed plants by modifying either the steam cycle or the cooling system alone. As we shall see in the case study demonstrations, different steam cycles not only have different capital and energy cost characteristics, they also have significant bearing on minimal cooling system costs.

In summary, a "steam cycle" is defined for our purposes to include the boiler-turbine-generator and related components of the total generating station, but excluding condenser and other circulating water system components. A given steam cycle

design is defined by its design values of thermal efficiency (heat rate), generating capacity, temperature of condensing steam (or turbine back pressure) and total heat load transferred to the cooling water at the condenser (condenser duty). The outputs of a given steam cycle analysis include a value for condenser duty which, along with vacuum, becomes a design condition for the cooling system evaluation (Step 3). In addition, the total incremental costs of design modifications with respect to the base case plant design—including capital investment, capacity loss penalties and fuel cost differentials—are calculated to be carried over to Step 4 where they are added to cooling system incremental costs calculated in Step 3.

Cooling System Design Evaluation

The portion of our overall procedure dealing with the evaluation of alternative cooling system design options has been automated in order to provide a capability for handling large numbers of potential design variations, including both different types of cooling system configurations as well as circulating water flowrate and temperature variations within each type. The computer program was written in FORTRAN IV language for operation on an IBM 360/65. The following discussion presents only a cursory overview of a rather complex engineering design sequence that is described in considerable detail in Appendix A of this report.

In our procedure, a "cooling system" is defined to include pipes (and/or tunnels or canals) used for intake and discharge, pumps, water treatment facilities (if any), a condenser, and, in some cases, a heat rejection component (e.g., a mechanical or natural draft cooling tower, or a cooling pond). Our definition of the cooling system specifically excludes components and considerations relevant to the ultimate disposal of the effluent in the receiving water. Thus, for example, the cooling system as we have defined it does not include components such as skimmer walls, underwater dams and diffusers.

As described above, all designs produced in a given computer run are mutually compatible and technically consistent with a given steam cycle design defined by design values for thermal efficiency capacity, condenser heat duty, and temperature of condensing steam.

Since our approach was developed primarily as a component of the sequential procedure previously introduced, it differs somewhat from conventional approaches to

cooling system design which have generally sized cooling system components to minimize the contribution of the cooling system to total power production cost, but without systematic treatment of thermal quality standards. Although our procedure can be used for this same purpose, its objectives are broader. Thus, as presently programmed, the procedure will produce a systematic array of alternative designs for any or all of three broadly defined cooling system flow configurations, namely, once-through, recirculation, and helper, with additional possible variations. Each such individual design yields different values of T_e and L_e . Once all designs are produced and their costs computed, a simple sorting routine reveals that subset of designs having effluents meeting given alternative thermal effluent standards T^* and L^* ; and, within that subset, the minimum cost design meeting each such standard is selected. The matrix of minimum cost designs with respect to all alternative L^* , T^* combinations, completes the cooling system design evaluation based on the given turbo-generator (steam cycle) design. This set of minimum cost designs is then carried to the next step in the procedure.

Combining Results for the Total Procedure

The last two steps in the procedure involve routine cost totaling and sorting operations.

First, the incremental cost of each steam cycle design option (determined in Step 2) is added to the incremental cooling system costs of its respective set of least cost cooling system design options evaluated in Step 4. This is simply a cost totaling step to combine the costs of the two separate best related components of each given plant design. If N separate steam cycle alternatives had been evaluated there would now be N different cost matrices representing the total incremental cost of the given steams.

At this point the procedure has developed and estimated combined costs for a "best" (least cost) system for each of the individual steam cycles for meeting all of the $m \times n$ alternative effluent standards, T^* , L^* . The final step is to determine, for any given effluent standard, which of the N individual, sub-optimized, steam cycle-cooling system combinations is the least cost overall plant design for meeting the given standard. This is accomplished by simple inspection across the N individual combined matrices. By repeating the process for all T^* , L^* , a single final matrix is developed, including the least-cost combined systems from among all N matrices.

II. PROPOSED-PLANT DEMONSTRATION—547 MW COAL-FIRED UNIT

Introduction

In this section we demonstrate the method of thermal effluent abatement cost analysis introduced in Section I by applying the method to a "real-world" situation involving a 547 mw coal-fired unit recently constructed on a river site in Pennsylvania. The general purposes of this case study are twofold. The first is to provide a reasonably detailed illustration of our methodology for abatement cost estimation, both with respect to our overall conceptual approach to the problem and also in terms of the specific capabilities of our cooling system design program. Hopefully, this will facilitate assessment of its relative merits and potentials for further development and application. The second purpose is to develop specific results pertaining to thermal abatement cost functions for generating stations of this characteristic size, type, and north-temperate regional location.

The section is divided into several parts, beginning with a brief description of the steam cycle and cooling system characteristics of the base case prototype plant. Following this we describe the eleven alternative steam cycle designs that were developed as variations on the base-case design and compare their salient technical design, thermal efficiency, and cost differences. We then present and analyze the cooling system design results developed from applying our cooling system design program to each of the twelve steam cycle alternatives. The separate results from the steam cycle modification and cooling system design analyses are then brought together in an analysis of least-cost combinations for achieving alternative effluent control levels. Following this we report briefly on results of a preliminary examination of two additional abatement options explored, namely, decreasing unit design capacity and reducing seasonal operating load.

The Base-case Plant Description

The base-case unit is a coal fired, single reheat generating unit located on a new site in Pennsylvania. The turbine is a four-flow, tandem compound machine with a guaranteed capacity rating of 547,180 kw gross at 3500 psig and 1000°F/1000°F steam conditions. The cycle uses seven extraction feedwater heaters.

The cooling system is a once-through type utilizing river water, with a dual pressure surface condenser having design back pressures of 1.72 and 2.15 inches of mercury (absolute pressure), respectively. The system is characterized by a design intake water flow rate of 253,000 gpm and a design intake temperature of 74°F. The design condenser duty is 2.24×10^9 Btu/hr, which leads to a circulating water temperature rise across the condenser of approximately 17°F.

The prototype unit was recently constructed, and as far as we could discern it was designed in the absence of thermal effluent constraints. In this sense, at least, it is believed to be reasonably consistent with our definition of a "privately optimal" design.

Steam Cycle Modifications

A total of twelve steam cycle cases were examined for the present study, including the base case design and eleven alternative variations, all of which were constrained to be capable of delivering the original unit capacity output of 547 mw. Although the number of alternatives was limited by budgetary considerations, the steam cycle modifications developed were explicitly structured in a manner to test the cost and performance trade-off implications of a number of combinations of condenser vacuum (turbine back pressure) and station heat rate.

The following pattern of design characteristics was specified as a basis for developing engineering and cost calculations for the individual plant designs. First, a series of alternative turbine back pressure levels were established as fixed design conditions. These were composed of three pairs—a "low," an "intermediate," and a "high"—selected to span virtually the full range of design back pressure that might be observed in practice for plants of this type and location, with special attention to the alternative cooling system types to be considered in our later analysis. Each pair, in turn, was comprised of a dual pressure and a single pressure variant, making for a total of six alternative back pressure configurations. Finally, for each of these six design back pressures it was determined that two alternative generating systems should be developed, one corresponding to a "most probable" design from an engineering-economic standpoint, and the other to a "best practical" design with respect to station heat rate (given the respective design back pressure constraints). The two steam cycles for each of the six back pressure conditions accounts for the total of 12 designs.

Description of Alternative Steam-Cycle Designs

As summarized in Table 1 below, steam cycle Cases 1-4 encompass the "low-back pressure" designs, Cases 5-8 the "intermediate" and Cases 9-12 the "high" back pressure cases. In one way or another, all of the designs numbered 2 through 12 constitute variations on Case 1, which is the original base case prototype unit. Thus, Case 2 represents the "high efficiency" (lowest practical heat rate) counterpart of Case 1; Case 3 involves modification of Case 1 for operation with a single pressure condenser designed to operate at a back pressure identical to Case 1's first-stage 1.72" vacuum; and Case 4 is the "high efficiency" counterpart of Case 3, utilizing the same, single-stage 1.72" condenser design condition. Cases 5-8 follow a similar pattern, with Case 5 being a redesign of Case 1 to operate on the higher design (dual) back pressures of 2.50"/3.05", Case 6 then being the high efficiency counterpart of 5, and Cases 7 and 8 being the two single back pressure variations for the intermediate 2.50" vacuum. The "high vacuum" Cases 9-12 once again repeat this pattern with respect to dual vs. single pressure design vacuums and "most probable" vs. "best practical" heat rate variations.*

The engineering design and cost calculations for these alternative steam cycles was performed by Jackson and Moreland Division of United Engineers and Constructors, Inc. In each instance the design engineer was instructed to utilize his best engineering-economic judgments to "reconstruct" the original plant design (exclusive of the circulating water system) on the basis of the revised back pressure and thermal efficiency assumptions, but holding constant the other background assumptions (generating capacity, fuel cost, fixed charge rate, etc.) of the original design. Specific kinds of technological modifications included adjusting boiler size, adding reheat cycles, modifying feedwater heaters, adjusting the size of the turbine exhaust end, and the like. Particular technical modifications for each design case are briefly summarized in Appendix C.

* Since there are a relatively large number of cases to be kept in mind, Table 1 constitutes a most important reference table for this entire section. The reader may also find it useful for later reference purposes to note that the even-numbered cases (2, 4, 6, etc.) always represent the "high efficiency" counterpart of the immediately preceding (odd-numbered) cases 1, 3, 5, etc..

TABLE 1

DESIGN CHARACTERISTICS AND INCREMENTAL INVESTMENT
COSTS FOR ALTERNATIVE STEAM CYCLE CASES DESIGNED
FOR THE 550 MW COAL-FIRED UNIT¹

Steam Cycle Case Number	Design Characteristics				Incremental Investment Costs ³	
	Condenser Vacuum (in. Hg abs.)	No. of Reheats	Station ² Heat Rate (Btu/Kwhr)	Condenser Duty (10 ⁹ Btu/hr)	Total (\$)	Total Annual ⁴ (\$/Year)
1	1.72/2.14	1	8664	2.241	Base	Base
2	1.72/2.14	2	8546	2.184	1,090,500	124,300
3	1.72	1	8639	2.228	-20,550	-2,340
4	1.72	2	8519	2.171	1,086,500	124,000
5	2.50/3.05	1	8827	2.317	-619,200	-70,600
6	2.50/3.05	2	8704	2.259	462,900	52,800
7	2.50	1	8784	2.297	-647,900	-73,900
8	2.50	2	8662	2.239	459,200	52,300
9	3.25/3.95	1	8941	2.372	-1,766,500	-201,000
10	3.25/3.95	2	8814	2.311	-674,400	-76,900
11	3.25	1	8914	2.358	-1,780,900	-203,000
12	3.25	2	8787	2.297	-673,845	-76,800

Notes:

¹All alternative cases designed for 547,180 Kw Guaranteed Output. Case 1 represents original prototype base-case unit. Calculations for all other cases were developed by Jackson and Moreland Division of United Engineers and Constructors, Inc., Boston, Massachusetts, based on specified design values for condenser vacuums, as given.

²Station heat rates = 1.145 x turbine heat rate values developed in design calculations.

³Incremental costs reflect modifications in turbine-generator, feedwater-heaters, and boiler size. Double reheat calculated as \$2.0232/Kw more than estimated costs for single reheat.

⁴Annual cost = total cost x annual fixed charge rate of 0.114.

For each design, calculations were developed for revised design turbine heat rates and condenser duties, and also the resulting total investment cost differential with respect to the individual unit (excluding cooling system). Results of these calculations are also summarized in Table 1. Due to budgeting limitations, these calculations were necessarily quite dependent on engineering judgment without benefit of highly detailed optimization studies. While no claim can thus be made for precise optimization from this approach, it is believed to provide a reasonable first approximation to the design and cost estimates that would actually have resulted from a detailed plant design study. In any event they appear consistent with the level of detail and accuracy required for the purpose of the present study.

The final steps in preparing the basic data for this segment of the study involved translating investment cost differentials into annual cost terms and estimating annual fuel cost differentials in relation to the base case design. Initial investment costs were charged at an annual rate of 11.5%. In all cases, the alternative units were assumed to operate for 8150 hours per year at an average 91.5 percent of the guarantee capacity level of 547,180 kw, thus producing a total energy output of 4.08×10^9 kw hr/year as the design basis for estimating annual fuel costs. Annual fuel cost increments with respect to the base case were then calculated as the product of this total energy output $\times$ the design's differential heat rate $\times$ a unit fuel cost of \$0.18 per million Btu. Resulting values for these annual cost increments are presented in Table 2.

The total incremental annual costs of alternative steam cycle modifications (the sum of incremental annual investment and fuel costs) becomes a key input to the final analysis, since they are later combined with similar estimates of cooling system costs to yield total overall cost calculations for the total combined systems. As previously noted, the design condenser vacuums and calculated condenser duties for the alternative steam cycles (also summarized in Table 2) serve as basic input design values for the cooling system design portion of the overall case study.

Analysis of Steam-Cycle Modification Results

Several features of the steam cycle modification results summarized in Tables 1 and 2 are worthy of special note in relation to the overall Case Study Analysis. We consider first the modifications in the physical characteristics (heat rate, condenser duty and back pressure) and then their direct incremental cost implications.

TABLE 2

INCREMENTAL COSTS OF STEAM CYCLE MODIFICATION FOR
TWELVE ALTERNATIVE DESIGNS FOR A 550 MW COAL-FIRED UNIT¹

Steam Cycle Case No.	Incremental Annual Costs (\$10 ³ /Year)			Inputs for Cooling System Design ²	
	(1) Investment ²	(2) Fuel Cost ³	(1)+(2) = (3) Total	Condenser Vacuum (in. Hg a's.)	Condenser Duty (10 ⁹ Btu/hr)
1	Base	Base	Base	1.72/2.14	2.241
2	124	-86	38	1.72/2.14	2.184
3	-2	-10	-12	1.72	2.228
4	124	-110	14	1.72	2.171
5	-71	120	49	2.50/3.05	2.317
6	53	30	83	2.50/3.05	2.259
7	-74	90	16	2.50	2.297
8	52	0	52	2.50	2.239
9	-201	210	9	3.25/3.95	2.372
10	-77	112	35	3.25/3.95	2.311
11	-203	190	-13	3.25	2.358
12	-77	90	13	3.25	2.297

Notes:

¹Excludes costs associated with cooling system.

²Originally presented in Table 1.

³Fuel cost increment calculated on the basis of differential heat rate x 4.08 x 10⁹ Kwhrs/year of operation x (0.18 x 10⁻⁶)\$/Btu fuel cost. The differential heat rate comparisons for all designs are made at the 91.5% capacity level for the respective designs which is the assumed average operating level for the unit.

Recall that the condenser vacuums were specified a priori as design conditions for development of the alternative steam cycle designs, and that the relatively large differences shown were chosen intentionally to test and observe their implications for other steam-cycle and cooling system design and cost characteristics. In light of these substantial variations in condenser vacuum—both with respect to the dual pressure and single pressure alternatives—one of the most interesting results is the relatively small changes produced in station heat rates and condenser duties compared to the base case values. Thus, in comparing the odd-numbered Cases 3, 5, 7, 9 and 11 (all of which were designed on the basis of a similar "most economic" heat rate assumption), none resulted in a differential heat rate relative to the base case of more than $\pm 3\%$ in spite of their large condenser vacuum differences. Thus, even for Cases 5 and 9, it was apparently possible to redesign a steam cycle against the higher design back pressure conditions in such a manner as to compensate substantially for the loss in thermal efficiency implicit in the higher design turbine back pressure; and also to produce the original 547 mw capacity output at no more than a 5% increase in total design condenser duty at the highest 3.25"/3.95" vacuum of Case 9.

Somewhat less surprising upon reflection are the relatively modest heat rate reductions observed for the paired comparisons of the "best practical" against the "most likely" designs (Case 2 vs. Case 1, Case 4 vs. Case 3, etc.). This is largely explicable by reference to the fact that the prototype design is itself already a very efficient unit at a design net heat rate of 8664 Btu/kw hr (corresponding to a net thermal efficiency of slightly over 39%). Thus the technical possibilities for further technical efficiency gains under the current state of the art are quite slim to begin with in this particular case study plant.

Essentially the same conclusion holds regarding the closely-related results for condenser duties, in which percentage reductions for the paired comparisons are also extremely modest. One of the principal reasons for producing the high efficiency designs was to explore potentials for reducing condenser duties and, thereby, reducing cooling system costs. As will be seen later, the cooling system implications of the small condenser duty changes achieved (e.g., comparing Case 2 vs. Case 1 or Case 4 vs. Case 3) are relatively insignificant.

Turning now to the direct cost implications of the steam cycle design modifications, it is important to reiterate that each of the alternative designs has been sized

for the original 547 mw capacity. Thus, for this case study there are no capacity-loss penalties associated explicitly with the steam-cycle modifications per se. The investment and fuel cost differentials indicated in Table 2 thus account for the total direct cost increments. Individually, both the investment and fuel cost categories show some relatively large changes—up to about \$200,000 each in annual terms for Cases 9 and 11 relative to the base case. However, for all but three of the alternatives, investment and fuel cost components have opposite signs, thus tending to offset one-another. The net effect is that the total cost changes attributable to steam cycle redesign turn out to be surprisingly modest overall, never exceeding \$83,000/year for Case 6, and ranging between \$52,000 and (-) \$13,000 in the remaining cases.

Cost comparisons for the high-efficiency designs with their lower efficiency counterparts indicate, as expected, that in every instance the higher efficiency case involves higher investment cost and lower fuel cost. This general result is consistent with the general proposition of a rising marginal cost of thermal efficiency widely observed for the technology [2]. In this particular study, the increase in investment cost of improved heat rate outweighs the resulting fuel cost saving by an order of \$25–40,000 per year (i.e., Case 2 has a total incremental cost of \$38,000 over Case 1, and Case 12 shows a difference of \$26,000 per year over Case 11). However, this particular result, as with many of the other total cost comparisons of this case study, is rather sensitive to the particular fixed charge rate and unit fuel price parameters utilized in calculating annual investment costs and total fuel costs.

Both the fixed charge rate of 11.5% and the fuel price of \$0.18 per million Btu's were values utilized in the original design studies for the prototype plant, and they were retained in our study to maintain consistency. Both values are known to be decidedly on the low side, being more representative of those prevalent in the mid-1960's rather than the early 1970's [5; 6]. Although we have not performed a detailed sensitivity analysis for these cost results, a general impression of possible changes can be based on the fact that both the fixed charge rate and fuel price enter the present steam-cycle cost calculations in a linear fashion. Thus, increasing both by the same proportion (say, 50%) would produce an increase in the sum of annual investment and fuel costs by the very same (in this example, 50%) proportion. On the other hand, if the parameters are adjusted by different proportionate increases, the total cost results could be either sig-

nificantly accentuated, reduced or even (in 8 of the 11 alternative cases) changed in sign.

One of the more interesting aspects of the incremental cost results, and one that is not immediately obvious from Table 2, is the pattern followed by total incremental cost as one moves from the low to the intermediate to the high back pressure cases, holding condenser type (single vs. dual) and relative efficiency variables constant. Thus, compare Cases 1, 5, and 9, for which total incremental cost of steam cycle modification first increases to \$49,000 per year at the 2.50"/3.05" vacuum and then decreases back to only \$9,000 per year at the 3.25"/3.95" vacuum of Case 9. This same pattern is repeated when one compares the dual-pressure/high-efficiency Cases 2, 6, and 10; or the single vacuum Cases 3, 7, 11; or the single-vacuum/high-efficiency Cases 4, 8, 12. In all instances, total cost first rises from the low to the intermediate design vacuum case, then falls back towards (or equal to) its original cost between the intermediate and the high vacuum cases.

This result was entirely unanticipated on our part, since a priori we would have expected total steam cycle cost to continue to increase in going from intermediate to high design vacuum conditions. The pattern observed is only partially a function of the relatively low fuel price used in our calculations, which if raised to, say, 33¢/million Btu's would have modified the results to the expected patterns. More significantly, the pattern is explained by the sharp reduction in initial capital investment costs for the turbo-generators installed for the high vacuum cases compared with the intermediate vacuum and low vacuum cases (see Table 1). This non-linear, decreasing pattern of investment costs thus tends to dominate the approximately linear increase in fuel costs in determining the observed pattern for total costs. It is not certain whether this result is typical and would be repeated for other turbine types and sizes, or whether it is particular to this case study type. However, it is a question worth further investigation, since as shown later in the analysis it can have important implications for selection of the least-cost combination of steam cycle and cooling systems.

At this point in the Case Study, twelve steam-cycle design options for a 547 mw coal-fired unit have been specified, and incremental costs have been calculated for the eleven alternatives to the base case unit. We now proceed to the next stage of analysis, in which each of the twelve sets of alternative condenser design conditions as sum-

marized in Table 2 are utilized as basic input parameters in the development of cooling system design options for each of the twelve steam cycle cases. The cooling system design results will then later be integrated with the incremental steam-cycle cost calculations in an analysis of least-cost system combinations for achieving alternative effluent control levels.

Cooling System Design Analysis

For each of the twelve steam cycle cases, in turn, one computer run of our cooling system design program was employed to develop and calculate the costs for an array of designs spanning technically feasible possibilities for each of the four alternative types of cooling system configurations discussed earlier, namely, the "once-through," "once-through-with-blending," "recirculation," and "helper" types. For both the recirculation and helper types, all three of our evaporative heat-rejector options—mechanical draft and natural draft towers and cooling ponds—were included in the present analysis. Recall that our definition of a "cooling system" includes intake and discharge piping, circulating water pumps, and condenser for all types of designs including those utilizing an evaporative heat rejector.

In addition to the condenser duty and vacuum input parameters which vary with each computer run (corresponding to changes in the respective steam-cycle cases), there are a large number of additional input parameters, all of which remain invariant for the entire set of twelve specified computer runs. These inputs include the hydrologic and climatologic design conditions and cost factors for the various cooling system components, as well as control options specifying design ranges and/or numbers of step-increments to be evaluated for particular design options. Definitions and functions for all cooling system program input variables are provided in general terms in Appendix A.

Insofar as possible, all cooling system input values for this particular case study were chosen either on the basis of published climatological information for the case study region or engineering and cost values employed by the engineering architect in the design of the prototype plant. To facilitate exposition, values for specific case-study inputs are listed in Appendix B, and will be introduced into the text selectively as necessary to explain observed results.

Number of Cooling System Designs Produced

The particular selection of input values for this study resulted in a total of approximately 2000 technically feasible cooling systems from the twelve computer runs as a basis for comparative cost and performance analysis. Table 3 provides a tally of the numbers of technically feasible schemes by cooling system type for each of the twelve steam cycle cases. As indicated, individual computer runs produced varying numbers, from a low of 12 designs each for Cases 3 and 4 up to a high of 345 individual designs for Case 9.

TABLE 3
TALLY OF NUMBER OF ALTERNATIVE COOLING SYSTEMS DESIGNED BY
COOLING SYSTEM DESIGN PROGRAM FOR EACH OF 12 ALTERNATIVE
STEAM-CYCLE CASES FOR 550 MW COAL-FIRED UNIT

Cooling System Type	Low Backpressure Cases				Medium Backpressure Cases				High Backpressure Cases			
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
Once-Through	5	5	3	3	8	8	6	4	10	10	8	8
Once-Blend	10	10	6	6	16	16	12	8	20	20	16	16
Recirc, M.D.	9	9	1	1	43	41	26	26	79	78	52	51
Recirc, N.D.	9	9	1	1	43	41	26	26	79	78	52	51
Recirc, Pond	9	9	1	1	43	41	26	26	79	78	52	51
Helper, M.D.	8	8	0	0	24	22	8	8	26	24	26	22
Helper, N.D.	8	8	0	0	24	22	8	8	26	24	26	22
Helper, Pond	8	8	0	0	24	22	8	8	26	24	26	22
Total No. Designs	66	66	12	12	225	213	120	114	345	336	258	243

The table makes it clear that more designs are produced as the design back pressure increases, and also that more designs are produced for the dual pressure Cases (1, 2; 5, 6; and 9, 10) than for the single pressure alternatives. In both instances the explanation lies in the fact that, other things equal, the range of technically feasible circulating water temperature rise across the condenser is constrained by the design condenser vacuum in all cases. Corresponding to standard engineering practice, the highest feasible hot water temperature in all cases is set at 5°F below the design temperature of condensing steam. The latter is a direct function of design back pressure, varying from a low of 96°F for the (1.72") single-vacuum, low-pressure Cases 3 and 4

up to a high of 125°F for the second-stage (3.95") vacuum of the dual-vacuum/high-pressure Cases 9 and 10. Thus, the low pressure cases have a much lower range of coolant temperature-rise within which to develop alternative designs, the number of which is determined on the basis of equal step-increments of either flowrate or temperature rise, depending on the design type in question.

Table 3 also indicates substantial differences in the number designs for different system types within given computer runs. This results partly from fundamental differences in technologies, partly from our intent to produce different degrees of "detail" in variation for different types of systems, and partly from differential effects of the above mentioned condenser temperature-rise constraint.

The Base-case Cooling System

Since the final cost analysis is made in terms of cost increments above or below the cost of a base-case, it is essential that the base case be well defined. In order to maintain consistency in design procedure and cost estimations for the cooling system portion of the case study, it was decided to develop the base-case cooling system design within our own computer modeling procedure rather than use the original prototype design and cost data. Thus, for purposes of this case study, the base-case cooling system was defined as the least-costly once-through system designed by the computer run for the base-case steam cycle (Case 1), selected independently of thermal effluent considerations.

The scheme thus developed was designed for a flowrate of 273,000 gpm and a corresponding effluent temperature of 90.4°F at the design intake water temperature of 74°F (or a temperature rise of 16.4°F). It should be noted that these characteristics quite closely approximate those of the once-through prototype design constructed for a flowrate of 253,000 gpm and a temperature rise of approximately 17°F. The basic similarity between the design produced by our computer model and that developed by the engineering architect of the prototype is an encouraging result.

As indicated below in Figure 1, the least-cost once-through scheme for steam-cycle Case 1 is calculated to have a total annual cost of \$279,200. This, then, is the base cooling system cost used in determining incremental annual costs (or savings) for all alternative schemes designed for all computer runs throughout this case study demonstration.

Selected Results for Individual Cooling System Types

Because certain design features and cost results for individual cooling system types are of some intrinsic interest in themselves, and also because an understanding of them is important in interpreting later cost comparisons, we present a brief overview of selected results for each of the major types examined. The results are presented in terms of total annual costs, with initial investment costs converted to annual terms using the same 11.4% fixed charge rate used in the steam cycle cost analysis. It should be kept in mind that these costs include only those attributable to cooling system components and do not at this point include cost differentials attributable to turbine and other steam cycle modification costs previously discussed.

Once-Through Systems. Figure 1 presents in graphic form the estimated total annual costs for all alternative once-through cooling systems designed for steam cycle Cases 1, 2, 3, 4, 5, 9, and 11, plotted as functions of design temperature-rise of circulating cooling water (and corresponding effluent temperature based on a design intake temperature of 74°F). Curves for the higher vacuum cases cover a broader temperature range for reasons previously discussed in connection with the numbers of designs produced. Note that all once-through designs were constrained by a maximum allowable intake flow of 550,000 gpm at a design intake temperature of 74°F.

Observe that in each case, total cost follows a characteristic "U-shaped" pattern in relation to temperature, first decreasing as design temperature rise (or effluent temperature) increases, reaching a minimum level, and then increasing for further temperature increases. This typical U-shaped relationship for individual cases is explained primarily on the basis of differential cost responses of major cooling system components—i.e., pipes and pumps on the one hand and condenser on the other—given that circulating water flow rate is inversely related to temperature rise for a fixed condenser duty (heat load). Thus, as temperature rise increases both pumping and piping costs decrease continuously due to the lower water flow required to remove the given heat load. At the same time, however, condenser surface area (and hence condenser cost) required to transfer the given heat load will continuously increase due to reduced heat transfer efficiency at the condenser as the effluent temperature approaches the fixed design temperature of the condensing steam (the latter being fixed in each case by the condenser vacuum design condition). The sum of these opposite effects produces the observed U-shaped cost pattern.

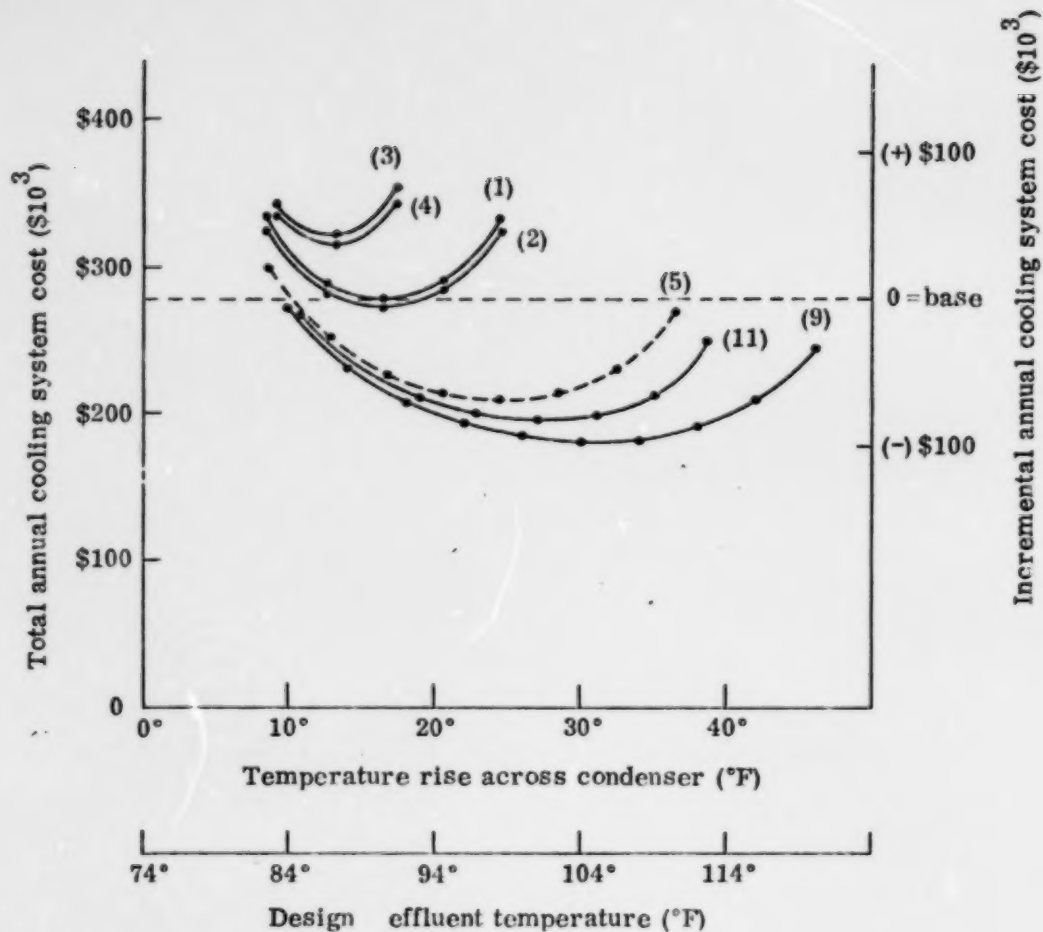


Figure 1. Comparisons of annual cooling system costs of once-through cooling systems, as functions of temperature rise or effluent temperature, for steam cycle design Cases 1, 2, 3, 4, 5, 9 and 11. 547 mw coal-fired unit.

Also of some interest are three additional features depicted in Figure 1. The first relates to the very narrow differential cost spread between Case 1 and its "high-efficiency" counterpart Case 2. This cost difference—on the order of \$5,000–7,000/year—between the two cases illustrates the relatively minor impact of the higher efficiency steam-cycle design in terms of reducing cooling system costs for once-through systems. A similar (i.e., less than \$10,000/year) comparative saving resulted for the high-efficiency counterparts of the other cases in this study. Note, secondly, that Case 3 (a single-pressure condenser) has higher costs than Case 1 for once-through cooling at all temperature levels. This, too, was a consistent pattern, single-pressure cases being more costly—although at lower temperature rises only just barely more costly—than the dual-pressure cases of comparable vacuum and thermal efficiency specifications. Finally, note the rather significant downward shift in the curve as between Case 1 and the higher dual-vacuum Case 5, for given temperatures, but that the decreasing cost effect of increasing vacuum is considerably diminished (particularly at the lower temperatures) for the shift observed between Case 5 and the highest vacuum Case 9.

It should be kept in mind that all the comparisons between cases illustrated in Figure 1 are on the basis of cost and temperature; and that, due to differences in heat load conditions among the cases, there are somewhat different cooling water flowrate values associated with any given temperature rise as between the different curves.

Once-Through-With-Blending Systems. For the present demonstration, two blending schemes were developed for every "standard" once-through design alternative, one utilizing the entire difference between the designed flow of its once-through "parent" scheme and the maximum allowable withdrawal of 550,000 gpm, and the other blending one-half of this allowable difference.

Blending provides a technical option to transform any once-through scheme of a given effluent temperature and flowrate into a lower-temperature but higher-flowrate system via the additional cost of required pipes and pumping components. Results of this case study suggest that most of the possible blending design alternatives for any given steam-cycle condition are redundant and non-economic from an effluent control standpoint, since it will usually be possible to achieve the same effluent temperature/

flowrate conditions at lower cost by an alternative (standard) once-through scheme involving a higher circulating water flow for the same vacuum conditions.

However, important exceptions occurred at the low end of the temperature-rise scale—the region of greatest significance from a thermal pollution control standpoint—in which some blending options proved somewhat less costly in achieving low effluent temperatures. This is illustrated in Figure 2 which presents comparative results from Case 9 of annual cost as a function of effluent temperature for the different cooling system types. In Figure 2 the "curve" for the once-through-with-blending scheme appears as the vertical line on the graph representing the 10 "maximum allowable" blending schemes, all of which had the same effluent temperature of 82.6°F (temperature rise of 8.6°F) and the maximum constrained effluent of 550,000 gpm.

Recirculating Systems. A traditional design optimization problem is the identification of the optimal range and approach for recirculation cooling systems involving both a condenser and a heat rejector. The cooling system design computer program is capable of approximating the "optimal" range and approach by systematically examining designs spanning feasible ranges of these variables and selecting the least costly design by inspection. To illustrate a typical array of designs produced by the program, Table 4 contains the total annual costs for all recirculation cooling systems generated in connection with steam-cycle design Case 9. Costs are given for each of the three types of heat rejector technologies examined in terms of both cooling range (and corresponding circulating flowrate) and approach.

It should be noted that all heat rejector schemes were designed against an ambient wet-bulb temperature condition of 73°F; and, where relevant, a design relative humidity of 50%. As discussed elsewhere, owing to limits imposed by available data in developing our tower cost functions, we restricted our designs by imposing a minimum range of 10° and a minimum approach of 8° rather than extrapolate values for extreme cases beyond these limits. For comparability, pond designs were similarly constrained in this study, although they need not have been, either on technical grounds or for reasons of data-availability. Due to somewhat arbitrary selection of control values for determining initial temperatures and step-increments (equal 2° steps in effluent temperature) the lowest approach value actually examined in this case was actually 8.9°F, or a design effluent water temperature of 81.9°F. This corresponds to

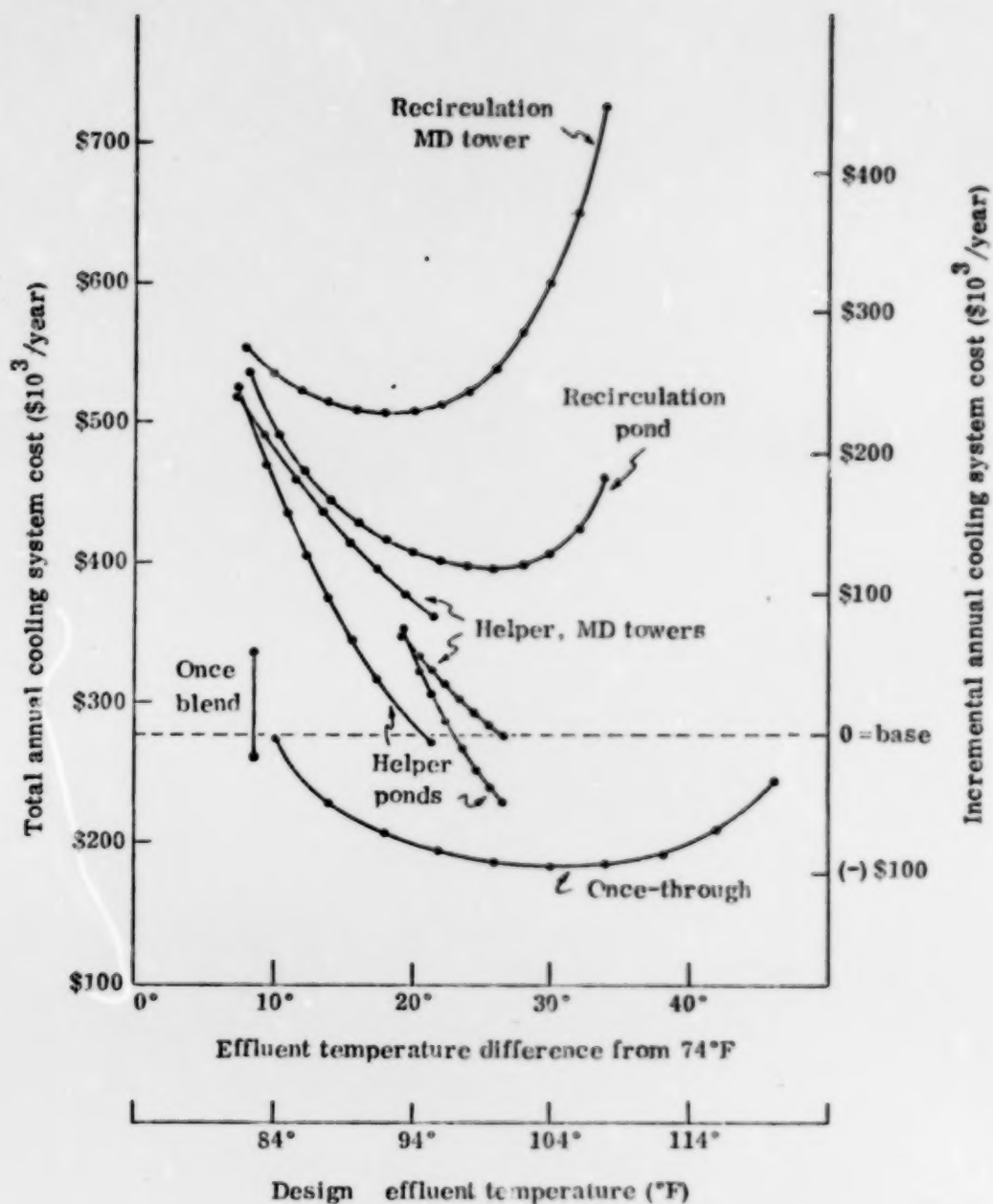


Figure 2. Annual cooling system costs as function of design effluent temperature, for alternative cooling system types. 547 mw coal-fired unit, steam cycle Case 9.

TABLE 4

TOTAL ANNUAL COSTS OF RECIRCULATION COOLING SYSTEMS
DESIGNED BY COMPUTER PROGRAM FOR STEAM CYCLE DESIGN
CASE NO. 9, 547 MW COAL-FIRED UNIT (\$10³/YEAR)¹

Approach (°F)	Cooling Range (°F) and Circulating Water Flow Rate (10 ³ GPM)						
	10° 474	11.2° 424	12.7° 374	14.6° 324	17.3° 274	21.1° 224	27.2° 174
34.9	734 854 462	727 835 483	n.f.d.	n.f.d.	n.f.d.	n.f.d.	n.f.d.
32.9	673 830 424	658 799 434	651 776 453	n.f.d.	n.f.d.	n.f.d.	n.f.d.
30.9	636 828 406	618 789 410	604 755 419	600 731 440	n.f.d.	n.f.d.	n.f.d.
28.9	613 840 398	593 796 399	576 754 403	565 718 413	570 699 441	n.f.d.	n.f.d.
26.9	601 863 398	581 813 *396	561 765 *396	546 721 401	539 686 415	n.f.d.	n.f.d.
24.9	599 896 403	577 842 400	556 788 398	537 737 398	524 691 404	531 *668 433	n.f.d.
22.9	604 941 414	581 881 409	558 822 404	537 764 401	519 709 402	513 *668 416	n.f.d.
20.9	616 999 430	592 934 423	567 868 417	544 802 411	522 739 407	*508 684 412	n.f.d.
18.9	635 1074 452	609 1002 444	582 929 435	556 855 427	531 783 419	510 716 417	517 679 444
16.9	660 1173 481	631 1092 471	603 1010 460	574 927 449	545 843 437	519 763 428	510 702 439
14.9	691 1306 519	660 1214 507	628 1120 493	596 1024 478	564 927 463	533 832 448	512 749 445
12.9	729 1489 570	694 1381 555	659 1271 538	623 1159 519	587 1045 498	551 930 476	521 823 462
10.9	772 1754 641	734 1624 621	694 1492 599	654 1356 574	614 1217 547	573 1076 517	536 939 490
8.9	821 2162 744	778 1999 717	735 1832 687	690 1661 654	614 1485 617	598 1305 576	554 1123 535

Notes:

¹ Numbers in each cell refer in order of descent to Mechanical Draft Tower, Natural Draft Tower, and Pond, respectively. *'s denote least costly systems of the respective types. n.f.d. = no feasible designs.

Design Conditions:

Condenser Duty = 2.327×10^9 Btu/Hr
Temp. of Condensing Steam = 124.96°F and 117.90°F
Wet bulb temp. = 73°F
Relative Humidity = 50%

a 7.9°F differential over the design ambient source water intake of 74°F. Tower and pond effluent (discharge) flows in all instances for Case 9 were calculated at less than 2,000 gpm.

The results confirm the expectation that the optimal (least-cost) range and approach for systems with different types of heat rejectors will likely be different. They also indicate degrees of cost variations within and between types. For all three types, the most costly designs were the ones with the lowest range and approach. For ponds and mechanical draft towers this "most difficult" design was between 60 and 80 percent more costly than the respective least costly design. However, for natural draft towers, the most costly design was on the order of 200 percent greater than the least costly design. The results also indicate that for comparable design conditions, ponds are less costly than mechanical draft towers, and natural draft towers are more costly than mechanical draft, on the basis of the costing parameters and functions used in this case study. However, it is evident that the cost differentials among the technologies vary significantly over different combinations of range and approach. Similar general results prevailed for all 12 cases explored.

The relative cost-effectiveness of the different heat rejector types is a complex issue, dependent on both differences in technical design features pertaining to heat transfer efficiency as well as costing parameters and functions used to evaluate both capital and operating costs of the three types. Throughout this case study, given our particular input values, the following general patterns for individual system cost components tended to prevail, by way of explaining the consistent total cost advantage of cooling ponds over the second-place mechanical draft tower schemes. For designs of the same range and approach conditions, the following conclusions can be drawn:

- (1) ponds had higher initial capital costs than MD towers for the basic heat-rejector component;
- (2) MD towers had higher initial cost for pump and fan components, but the difference was not enough to offset the pond's initial capital cost disadvantage;
- (3) MD towers had considerably higher annual operating energy cost, water treatment cost, maintenance cost, and capability penalty cost (resulting from higher fan and pump energy requirement), the sum of these four current cost items being

more than enough in virtually all instances to more than offset the pond's initial (net) investment cost disadvantage, and result in higher total annual cost for the tower designs.

One objective in bringing out all of this detail is simply to demonstrate that, in the present study at least, operating costs played a decisive factor in leading to the selection of the pond over tower designs, both for recirculation and helper systems. Clearly, the designs produced in our study are close enough in total cost to suggest that modest changes in our procedures for estimating operating costs could have led to the mechanical draft towers being generally less costly than the pond schemes. Just as clearly, the closeness of the total costs between these two heat rejector types suggests that modest changes in the parameters of the cost functions for the heat rejectors or an increased fixed charge rate could also have produced such a reversal. While these points are reasonably well known, we feel they bear emphasis since they help to put our numerical results in perspective. While we did not undertake a formal "sensitivity analysis," the above discussion suggests that such an analysis would yield interesting results.

In Figure 2, the least-costly mechanical draft tower and cooling pond designs for each alternative approach temperature (as presented in Table 4) are plotted in terms of total cost relative to discharge temperature. In both cases costs first decrease with increased approach (effluent) temperature, but then begin to rise at an increasing rate beyond a minimum cost level. Since these curves reflect the influence of changes in both approach and range, with total heat load held constant, they should not be confused with the standard cooling tower design curves that show cost (or "degree of difficulty") decreasing continuously with increase in design approach temperature for a fixed cooling range. Recall also that these costs include variations in condenser area, pumps, and piping as well as the heat rejector, per se.

Helper Systems. Helper designs—i.e., schemes interposing an evaporative heat rejector between a once-through condenser design and the cooling effluent discharge point—were constructed on the basis of the same climatological design conditions used for the recirculation systems. Basically, these systems provide an intermediate technical option between once-through and recirculation types for achieving effluent controls, in that they provide for the low effluent temperature potentials of the recirculation schemes but at the high flowrate values comparable to once-through designs.

Reference to Figure 2 provides a partial illustration of this potential, showing Case 9 least cost results in relation to effluent temperature, compared with other cooling system types, for both mechanical draft towers and ponds utilized in the helper mode. Two sets of curves are presented for helper designs; the curves to the left being instances where the entire once-through condenser flow is directed through the heat rejector prior to discharge, and those to the right being variations in which only one half the condenser outlet flow is passed through the heat rejector and then blended back with the other half prior to discharge.

Results in Figure 2 show that, for given effluent conditions, helper ponds are generally less costly than mechanical draft towers for our case study input values, a result previously noted and explained in connection with recirculation schemes. The notable exception to this general result for helper designs occurs at the very low temperature extreme where total costs become virtually identical, with a mechanical draft tower actually being less costly than a pond in one instance.

It should be noted once again, that in comparing costs of different cooling system types in Figure 2, comparison is made on the basis of effluent temperature only, and that flowrates associated with these temperatures are quite different as between recirculation, helper, and once-through designs.

Cooling System Cost Implications of Steam Cycle Redesign

Recall that a principal reason for examining alternative steam cycle designs is to determine the influence of variations in steam cycle design for costs of potential cooling system options, and that each of the twelve cooling system design computer runs represents an experiment along these lines. Although space prohibits detailed comparisons for all basic cooling system types across all twelve design runs, a brief summary of some of the key results will provide essential background for our later total plant cost comparisons. In general, we are interested in the implications for cooling system costs of three types of changes: (1) increased thermal efficiency (reduced heat rate); (2) substituting single-pressure for dual-pressure condensers; and (3) increasing the unit's design back pressure.

We have already analyzed the effects of each of these factors in some detail with respect to once-through cooling systems. Recalling our discussion in connection with Figure 1, we found, comparing the "high-efficiency" (even numbered) cases against

their otherwise-similar "most probable efficiency" counterparts, that the small changes in heat rates and condenser duties possible for our case study plant resulted (not unexpectedly) in only nominal cost savings for once-through cooling systems—on the order of \$5,000–7,000 per year for designs of comparable effluent flowrate and/or temperature characteristics. The same kind of comparison for other cooling system types reveals that the cost change is also in the proper direction—i.e., lower cooling system costs for the more efficient designs with their reduced condenser duties—and also quite modest. Other things equal, as a rather crude general rule, the cost savings for the various cooling system types tended to be on the order of 2–2.5%, or roughly equivalent to the percentage improvement in heat rate resulting from the steam cycle modification. Thus, in the case of the higher cost recirculation designs the absolute size of cooling system cost savings was a more substantial \$15,000–20,000 per year for optimal designs based on the improved efficiency steam cycles.

Again recalling our findings for once-through systems, it was seen that substituting single-pressure for dual-pressure condenser conditions resulted in a cooling system cost increase. However, further examination reveals that the size of this cost increase tends to be exponentially related to design flowrate or temperature rise. Thus, at very high flowrates (lowest temperature rises) cost differences between the otherwise similar cooling systems were virtually negligible between the single-pressure and dual-pressure cases; but as design flowrate decreased (temperature rise increased) the single-pressure designs become increasingly more costly as the influence of the higher second-stage back pressure increasingly overwhelms the influence of the lower condenser duty of the single-pressure design. Generally similar results tended to prevail for other cooling system types, with single-pressure designs invariably implying higher cost cooling systems compared with dual-pressure alternatives. This cost disadvantage was greatest for the higher-cost tower and pond systems than for once-through designs. However, the magnitude of these cost differences tended to diminish as one progressed from the lower to the higher back pressure cases. Thus, for example, we find a cost difference between Case 3 and Case 1 of some \$200,000 per year in favor of Case 1 for the least costly recirculation designs; but this diminishes to about \$45,000 per year disadvantage for Case 11 as compared to Case 9. Similarly, for once-through systems the cost disadvantage for the least-costly single-pressure

scheme diminishes from \$43,000 when comparing Case 3 to Case 1 to only \$14,000 as between Cases 11 and 9.

This takes us to the final set of comparisons of present interest, those pertaining to an increase in design back pressures from low to intermediate to high. Here we have seen from previous discussion that cooling system costs fall, although at a decreasing rate, over this span of alternative condenser vacuums. And it is here that we find our most dramatic cost differences in absolute terms. Comparing Cases 1, 5, and 9 with respect to once-through systems (Figure 1) at the lowest temperature levels, we see a cost decrease of \$35,000 per year from Case 1 to Case 5, but a negligible change from Case 5 to 9. On the other hand, comparing the minimum cost points on the three curves (irrespective of effluent temperatures) indicates a decrease of \$77,000 between Cases 1 and 5 and a further decrease of \$29,000 per year between Cases 5 and 9.

However, the largest absolute cost advantages for the high back pressure cooling schemes occur for the recirculation designs. Thus, across Cases 1, 5, and 9, respective least-cost recirculation pond schemes decrease in total annual cost from \$720,000 to \$483,000 to \$396,000, an overall reduction of \$324,000 in cooling system costs attributable to adoption of the 3.25"/3.95" as opposed to the original 1.72"/2.14" vacuum. For the best mechanical draft towers designed for Cases 1 and 9 the comparable annual cost reduction was about \$300,000 per year.

In summary, other things equal, the cost of cooling systems in this case study are slightly lower for the higher efficiency (lower heat rate) cases, somewhat higher for single-pressure as opposed to dual-pressure condenser designs, and, most significantly, lower for the higher condenser pressure cases. Thus, as shown in Table 5, comparisons among all twelve cases would show the highest cooling system costs for Case 3 (single-low pressure/normal efficiency conditions) and the lowest cooling system costs for Case 10 (dual-high-pressure/highest practical efficiency design conditions). These general conclusions hold for all cooling system types examined, but in absolute terms the most significant cost differences between cases are associated with the schemes involving cooling towers or ponds. It is of some interest to note with respect to Table 5 that all steam cycle alternatives with the exceptions of Cases 3 and 4 imply lower costs of comparable cooling systems than the original base-case plant.

TABLE 5
RELATIVE RANKINGS OF TWELVE ALTERNATIVE
STEAM-CYCLE/CONDENSER DESIGN CASES IN TERMS OF
ANNUAL COOLING SYSTEM COSTS FOR COMPARABLE
COOLING SYSTEM TYPES AND EFFLUENT CHARACTERISTICS

Condenser Design Conditions	Design Case Numbers	Relative Cooling System Cost Rank
<u>Low Pressure Cases</u>		
Single Pressure		
Normal efficiency	3	Highest
High efficiency	4	2nd
Dual Pressure		
Normal efficiency	1	3rd
High efficiency	2	4th
<u>Intermediate Pressure Cases</u>		
Single Pressure		
Normal efficiency	7	5th
High efficiency	8	6th
Dual Pressure		
Normal efficiency	5	7th
High efficiency	6	8th
<u>High Pressure Cases</u>		
Single Pressure		
Normal efficiency	11	9th
High efficiency	12	10th
Dual Pressure		
Normal efficiency	9	11th
High efficiency	10	Lowest

Before combining the cost and effluent performance results of the cooling system design analysis with the cost differences on the steam-cycle side, we need only introduce the "effluent standards" format in which the final results are to be presented.

Effluent Standards and Pattern of Least-Cost Cooling Systems

One feature of the cooling system design program is a sorting subroutine to select from among all schemes designed in a given production run the least-cost scheme capable of satisfying each of a predetermined set of effluent flowrate (L^*) and

temperature (T^*) constraints. For the present case-study demonstration, six alternative values of L^* were selected (ranging in value from a maximum of 550,000 gpm in equal increments of 100,000 gpm down to a minimum value of 50,000 gpm) and 11 values of T^* (ranging from 114°F in equal 4° increments down to 74°F). Each of these 66 possible combinations of L^* and T^* can be thought of as an alternative, two-dimensional "effluent standard." Arranged in matrix form in Table 6, they become an ordered array of standards that become increasingly restrictive the lower the value of L^* and/or the lower the value of T^* .

Both the maximum values and the sizes of increments for L^* and T^* can be set at the discretion of the analyst. In this case they were selected to provide a maximum relevant range for both variables (550,000 gpm was the maximum allowable intake value, 114°F was close to the maximum possible effluent temperature and 74°F was the ambient intake temperature), and to provide sufficient detail to make for "interesting" results without becoming computationally burdensome.

The least-cost cooling systems with design effluent flowrates and temperatures at or below each of the alternative sets of maximum allowable L^* and T^* effluent standards were thus determined within each of the twelve separate computer runs. The general pattern of cooling system types that emerged as the least-costly schemes for given combinations of L^* and T^* tended to be quite consistent across all twelve cases. In other words, the same cooling system types tended to dominate particular regions of the effluent standards matrix in many or all cases, as displayed in Table 6.

The solid lines in Table 6 isolate two regions in which the least-cost cooling system type was the same in each of the 12 runs: a high-flowrate, high-temperature region at the upper left where once-through designs were always least costly, and the lowest effluent flowrate region at the extreme right where recirculation systems become the only technically feasible designs. The intermediate region between the two lines is one where there was some variation in system type between the runs; but even here a pattern is evident, as the once-through schemes give way to blending and helper designs in the high-flowrate, low-temperature corner, and to recirculation systems at low-flowrate, high-temperature corner. Helper systems dominate most of the lowest temperature region, except at the very lowest flowrates, and much of the remaining intermediate region between the once-through and recirculation designs.

TABLE 6

GENERAL PATTERN OF COOLING SYSTEM TYPES DETERMINED TO BE LEAST
COSTLY FOR ATTAINING ALTERNATIVE THERMAL EFFLUENT STANDARDS
547 MW COAL-FIRED UNIT

Effluent Temperature T* (°F)	Effluent Flowrate, L* (10 ³ gpm)						
	550	450	350	250	150	50	
114°	Once-Through Systems Least Costly in All 12 Cases			0	0;R	Recirculation Ponds Least Costly in All 12 Cases	
110°				0	0;R		
106°				0	H;R		
102°				0	H;R		
98°				0	H;R		
94°				H	H;R		
90°	0		H	H;R			
86°	0;B	0;B	H	H	H;R		
82°	H	H	H	H	R		
78°	No Feasible Cooling Systems in Any of the 12 Cases						
74°							

Notes: Each entry indicates the most frequently occurring least cost cooling system type(s) among the 12 case runs. Unless otherwise noted, a single entry indicates selection in 8 or more of the 12 runs. Where two types are indicated, each of the two types was least-costly between 5 and 7 times out of 12. Symbols are defined as follows:

O = Once-Through
B = Once-Through-With-Blending
H = Helper Pond
R = Recirculation Pond

It should be noted in interpreting results such as these that they are partly due to cost differences, and partly to differences in technical feasibility per se. Thus, for example, recirculation systems in any of the runs had no competition at the very low flowrates, except among alternative recirculation schemes. Similarly, no once-through systems were designed in any of the runs with effluent temperatures as low as 82°F, although at least one in any given run approached this constraint quite closely.

In a few instances, somewhat arbitrary selection of program control variables has the effect of excluding schemes from being designed that might later prove least-costly in achieving a particular standard. For example, alternative blending rates for the once-through-with-blending schemes could have altered certain results. Likewise, setting slightly different effluent standard constraint increments would have altered the matrix patterns for some runs, particularly in the intermediate regions where shifts in the dominance of scheme types occurs most often. However, our intention has not been to design all possible schemes or delineate precise absolute values, but rather to demonstrate our general methodology and identify broad results and issues that appear worthy of further exploration.

As an indication of the absolute magnitudes of cooling system cost differentials compared to the base case (once-through) cooling system design, for schemes required to achieve various effluent standards, Tables 7 and 8 present the incremental cooling system cost matrices for design Cases 3 and 9, which encompass the overall range from highest to lowest cost cooling systems as previously discussed. Similar matrices for the other ten cases would generally show a rank ordering of incremental cost for given L^* and T^* effluent constraints between these extremes according to their relative position given in Table 5. Note that in each instance costs in Tables 7 and 8 are stated as total annual cost increments above or below the \$279,000 per year cost of the base case cooling system, previously defined as the least-cost once-through scheme designed for the Case 1 steam cycle.

Two particular features of the results presented in Tables 7 and 8 are worth special attention. The first pertains to the absolute values of incremental cooling system costs required in achieving the various standards. These vary between \$43,000 and \$643,000 for the extreme high cost Case 3 cooling system alternatives down to a range of (-) \$96,000 to \$255,000 for the generally least costly designs of Case 9. The comparable

TABLE 7

MATRIX OF INCREMENTAL ANNUAL COOLING SYSTEM COSTS FOR
LEAST COST SYSTEMS RELATIVE TO EFFLUENT STANDARDS
547 MW COAL FIRED UNIT, RUN NO. 3 (\$10³)

Effluent Temperature Standards T* (°F)	Effluent Flowrate Standards, L* (10 ³ gpm)					
	550	450	350	250	150	50
110°	0 \$43	0 \$43	0 \$43	R \$643	R \$643	R \$643
106°	0 \$43	0 \$43	0 \$43	R \$643	R \$643	R \$643
102°	0 \$43	0 \$43	0 \$43	R \$643	R \$643	R \$643
98°	0 \$43	0 \$43	0 \$43	R \$643	R \$643	R \$643
94°	0 \$43	0 \$43	0 \$43	R \$643	R \$643	R \$643
90°	0 \$43	0 \$43	0 \$43	R \$643	R \$643	R \$643
86°	0 \$62	B \$79	R \$643	R \$643	R \$643	R \$643
82°	R \$643	R \$643	R \$643	R \$643	R \$643	R \$643
78°	No Feasible Cooling Systems for T* ≤ 78°					

TABLE 8

MATRIX OF INCREMENTAL ANNUAL COOLING SYSTEM COSTS FOR
LEAST COST-SYSTEMS RELATIVE TO EFFLUENT STANDARDS
547 MW COAL-FIRED UNIT, RUN NO. 9 (\$10³)

Effluent Temperature Standards T* (°F)	Effluent Flowrate Standards, L* (10 ³ gpm)					
	550	450	350	250	150	50
110°	0 -\$96	0 -\$96	0 -\$96	0 -\$96	H -\$93	R \$116
106°	0 -\$96	0 -\$96	0 -\$96	0 -\$96	H -\$50	R \$116
102°	0 -\$94	0 -\$94	0 -\$94	0 -\$94	H -\$50	R \$116
98°	0 -\$87	0 -\$87	0 -\$87	0 -\$87	H -\$13	R \$118
94°	0 -\$73	0 -\$73	0 -\$73	H -\$3	H \$16	R \$128
90°	0 -\$52	0 -\$52	0 -\$52	H \$67	H \$67	R \$149
86°	B -\$30	B -\$30	H \$132	H \$132	H \$137	R \$201
82°	H \$240	H \$240	H \$240	H \$240	H \$240	R \$255
78°	No Feasible Systems Designed for T* ≤ 78°F					

range for Case 1, the base-case is from \$0.0 (the base once-through design) up to \$153,000 for a recirculation scheme. Compared with the incremental costs of steam-cycle modification per se—i.e., ranging from (-) \$15,000 to \$83,000 as shown previously in Table 2—incremental cooling system costs often tend to be as much an order of magnitude or more higher, especially for the low effluent temperature and flowrate schemes. This will be important to our later analysis of the combined systems.

A second feature of the results shown in the matrices is one commonly noted in the engineering-economics literature of pollution control, and one which conforms to the classical economic "law of increasing cost." That is, achieving increasingly higher degrees of effluent control involves rising costs and that, beyond some point, costs rise disproportionately for additional increments of effluent control. This can be seen in the tables presented, both with respect to decreases in effluent flowrate (reading across virtually any given row of the matrix representing a given T^*), and also with respect to incremental temperature decreases (reading down any given effluent flow column). The increasing marginal cost of effluent control is even more pronounced if it is required that both flowrate and effluent temperature be simultaneously reduced.

We proceed next to an integration of the cooling system design and cost results with those of the steam-cycle modification experiments, per se, as previously presented.

Synthesis of Steam-Cycle and Cooling System Design Results

Thus far, the cost results from the steam-cycle modification experiments and the cooling system design components of the overall procedure have been discussed separately. The next step in the procedure is to combine the steam-cycle incremental annual cost (SIAC) calculations for each of the twelve alternative cases (as summarized in Table 2) with their corresponding cooling system incremental annual cost (CIAC) matrices (as illustrated in Tables 7 and 8) to provide total incremental annual cost (TIAC) estimates for the alternative combinations of steam-cycle and cooling system components.

This step is accomplished routinely within the cooling system design computer run for each case simply by adding the appropriate SIAC value (a constant entered as a program input for each run) into the matrix of CIAC values developed by the computer

for the particular design case. Each element of the resulting matrix thus consists of the sum of the incremental cost of the steam cycle modification in question plus the incremental cost of its least-costly cooling system option for achieving a given L^* , T^* effluent standard. Each of these twelve matrices, based on ordered pairs of L^* , T^* values, can now be thought of as providing an approximate solution to the problem of minimizing thermal effluent abatement costs as a function of the two-dimensional effluent standard for the steam cycle alternative in question relative to the base case steam cycle.

The final step in the procedure is to select from among all twelve TIAC matrices the least-cost combination of steam cycle and cooling system for each alternative effluent control level. The matrix composed of these final minimum TIAC values thus constitutes an overall minimum cost solution; that is, it constitutes the thermal abatement cost function for the case study as a whole, integrating all twelve steam cycle possibilities and their corresponding cooling system options.

Analysis of Combined Case Study

The integrated results for the case study as a whole are presented in Table 9, following the same matrix format as Tables 7 and 8, with the steam cycle case numbers indicated in parentheses next to the symbols designating the cooling system types. It is evident that the two high vacuum condenser design Cases 9 and 11 dominate the final outcome, with Case 11 proving least costly for most effluent standards except for the lowest flowrate where the recirculation ponds for Case 9 prevail and the next lowest (150,000 gpm) flowrate where various types from Cases 9 and 12 appear. The only exception to this pattern is for the 550,000 gpm, 82° standard where a Case 2 blending scheme proves the low cost choice.

Most of the final choices indicated in Table 9 can be readily explained by reference to our earlier discussions of steam cycle modification and cooling system design cost differences among cases as summarized in earlier tables. Thus, recall (Table 5) that the high pressure Cases 9 through 12 were the four lowest ranked in terms of relative cooling system cost, and that cooling system cost differences among these four cases were generally very small. This, together with the fact that Case 11 had the least costly steam cycle of all twelve cases (Table 2)—actually a \$13,000 per year saving compared to the base case—explains its general dominance of the final comparisons.

TABLE 9

INTEGRATED MATRIX OF TOTAL INCREMENTAL ANNUAL COSTS FOR LEAST
COST COMBINED SYSTEMS RELATIVE TO EFFLUENT STANDARDS
547 MW COAL-FIRED UNIT, RUN NOS. 1-12 (\$10³)

Effluent Temperature Standards T* (°F)	Effluent Flowrate Standards, L* (10 ³ gpm)					
	550	450	350	250	150	50
110°	O (11) -\$95	O (11) -\$95	O (11) -\$95	O (11) -\$95	O (9) -\$84	R (9) \$126
106°	O (11) -\$95	O (11) -\$95	O (11) -\$95	O (11) -\$95	O (12) -\$68	R (9) \$126
102°	O (11) -\$95	O (11) -\$95	O (11) -\$95	O (11) -\$95	H (11) -\$42	R (9) \$126
98°	O (11) -\$93	O (11) -\$93	O (11) -\$93	O (11) -\$93	H (11) -\$5	R (9) \$127
94°	O (11) -\$83	O (11) -\$83	O (11) -\$83	O (11) -\$83	H (11) \$23	R (9) \$137
90°	O (11) -\$65	O (11) -\$65	O (11) -\$65	H (11) \$72	H (9,11) \$76	R (9) \$158
86°	B (11) -\$38	B (11) -\$38	H (11) \$131	H (11) \$131	H (9,11) \$146	R (9) \$192
82°	B (2) \$96	H*(11) \$248	H*(11) \$248	H*(11) \$248	H*(11) \$248	R (9) \$264
78°	No Feasible Systems Designed for T* ≤ 78°F					

Although Case 9 had a somewhat higher SIAC (by \$22,000) than Case 11, its relatively greater cost advantage with respect to recirculation schemes enabled it to be the preferred low-flowrate combination.

What appears at first glance to be an anomalous result in the choice of the Case 2 blending scheme for the 550,000 gpm, 82° effluent standard, turns out on further examination to be an interesting instance in which a steam cycle redesign for slightly higher thermal efficiency (and consequent lower condenser duty) made the slight difference in technical feasibility required to enable the 82° temperature standard to be met by a once-through-with-blending scheme rather than a halper scheme. Since blending is generally a much lower cost technology than helper schemes requiring ponds or towers, choices between the two types in the high flowrate, low temperature region of the matrix typically depend on whether or not the blending schemes can achieve a low enough effluent temperature to be technically competitive. The only instances in which blending schemes were technically capable of satisfying the 82° T* constraint under the design conditions of this study were for the two highest thermal efficiency (lowest condenser duty) Cases 2 and 4, with the Case 2 combination proving the less costly of the two options by only a negligible amount. However, the next closest substitute for these blending schemes for achieving the 550,000 gpm, 82° standard was the \$248,000 per year helper scheme of Case 11, which was about \$150,000 per year more costly.

It should also be carefully noted that this particular result is especially sensitive to a number of the design parameters of the study. Thus, had the condenser duties for all cases generally been only 2-5% lower (owing, say, to a slightly lower station generating capacity) or had our maximum design allowable intake flowrate been only modestly higher, any of a number of design cases could have produced technically feasible, relatively low-cost blending schemes capable of meeting the 82° standard. Alternatively, had the 82° standard itself been raised by as little as 0.5° to 1.0°, many other cases would have yielded competitive blending schemes. The results here underscore the conclusion that very small differences in design conditions or choice of effluent standard can make very substantial differences in the design selection and total annual cost for achieving fixed effluent standards, especially at the margin between technically competing alternative technologies where the cost functions are often highly discontinuous.

In terms of absolute costs of meeting alternative standards, Table 9 reveals that the most stringent effluent standard achievable in the present case study ($L^* = 50,000$ gpm, $T^* = 82^\circ\text{F}$) could be achieved by a recirculation pond at a total incremental cost of \$264,000 per year. It also indicates that only slightly less restrictive standards than these can usually be met at a cost typically on the order of only one-half to two-thirds this value. Furthermore, some relatively large reductions in either effluent temperature or flowrate as compared to the base case design values of $T = 90.4^\circ\text{F}$, $L = 253,000$ gpm could be achieved in this study by various combinations of steam cycle and cooling system redesign at only nominal incremental cost.

However, some of the absolute value results, especially those relating to once-through-and-blending designs may be called into question by the appearance of negative total incremental cost values, indicating that many of our designs are, for one reason or another, less costly than the base case design. We return to this issue below after first pointing up some additional relative cost comparisons between our final design selections and some of the alternative higher-cost case results.

High back pressure cases dominate the final results due to the fact that their incremental steam-cycle modification costs are quite nominal and their cooling system costs are generally considerably lower than either the low or intermediate back pressure cases. Thus, if we were to compare Table 9's results, which integrate all twelve cases, with those of a similar matrix integrating only the first four (low back pressure) cases, we would find that the "best" of the four low back pressure TIAC's are from about \$90,000 to \$310,000 more costly than the high back pressure results dominating Table 9. A similar comparison with a matrix integrating the intermediate vacuum designs (Cases 5, 6, 7 and 8) would show that the least costly of these four intermediate cases runs from \$50,000 to \$125,000 more costly (after accounting for their higher costs of both steam cycle and cooling systems) than the dominant high pressure cases in Table 9. Essentially, these comparisons measure the net payoff, in terms of annual cost savings, of considering high vacuum redesigns as an element in the effluent modification program for the present case study plant.

On the other hand, redesign for greater thermal efficiency and lower condenser duty did not, for the most part, result in any net savings as an abatement measure. Generally, the cooling system cost savings associated with the high-efficiency designs

(when compared with their normal-efficiency counterparts) were insufficient to offset their relative steam-cycle investment cost disadvantage; however, the net resulting differences between them were always relatively small—on the order of \$20,000 to \$40,000. The only exception to this rule occurred in a few instances (as discussed above in relation to the 550,000 gpm, 82°F effluent standard) where the lower condenser duty allowed for substitution of a lower cost cooling system technology in meeting a particular temperature standard. As we shall see later on, a change in the unit fuel cost parameter can completely reverse the net cost disadvantage of the high-efficiency designs.

Substituting single vacuum for dual vacuum condensers did sometimes pay off with a net cost reduction in achieving effluent controls (for example, as seen in the placement of Case 11 in Table 9); although, again, the net cost differences between single and dual pressure designs of the same relative efficiency and vacuum level were relatively modest.

Negative Incremental Cost Paradox

In a number of instances the final results of integrating steam cycle and cooling system incremental costs have yielded negative incremental cost values. Indeed, this occurred quite consistently among the once-through designs for the intermediate and high back pressure Cases 5–12, and also appeared in some of the partial helper schemes and blending designs of the high pressure cases. This came as an unexpected result, since one might have supposed that the base-case steam cycle and its own least cost (base case) once-through cooling system should have proven the most economic overall design combination, or at least very close to it. In fact, the base case turbine-condenser combination turned out to be among the more costly of the cases considered. The implication here is that either (1) the original prototype unit was not actually the most appropriate design from the standpoint of minimizing generating costs, given the design conditions specified, or (2) there is a systematic error in our present cost-estimating approach, or (3) some combination of the two.

Our own negative incremental cost results can be explained on the basis of the following combination of factors, each of which has previously been discussed individually. The first factor is the relatively large annual saving in cooling system cost attributable to increases in design back pressure (see Figure 1). The general effect

here is soundly based on thermodynamic engineering principle: i.e., that larger differences between steam condensing temperature and (given) cooling water intake temperature result in more efficient heat transfer and thereby require less circulating water flow and/or less condenser surface area for transferring a given amount of heat. Both of these effects, in turn, result in lower cooling system costs for the higher design back pressures. A possibility is that our particular methods of costing these effects may possibly have tended to overstate the negative cost results; however, we have no present reason to think this to be the case, and can only suggest it as a subject for future verification research.

The second factor is the relatively low fuel cost penalties associated with the higher design back pressures. This results, in part, from the smaller-than-expected thermal efficiency (heat rate) disadvantages for the higher pressure cases (see Table 1) and, in part, from the unit fuel cost factor ($\$0.18/10^6$ Btu) used in assessing the economic value of the penalties. The heat rates were double-checked by our engineering design consultant, and we return to the question of fuel price later on. Had the positive fuel cost penalties for Cases 5-12 been higher, they would have more completely offset the negative cost influence of the cooling system components.

The third factor is the quite-unanticipated, substantial decrease in turbine investment costs at the higher back pressures (holding guarantee output constant at 547 mw). Although other steam-cycle components (such as boiler cost) increased, turbine cost decrease was such as to swamp the other initial costs and lead to the large negative investment cost factor shown in Table 2 for Cases 5, 7, and 9-12. These, in turn, were substantial enough to largely offset (and for Case 11 to more than offset) the positive influence of the respective fuel cost penalties for these six intermediate and high-pressure design alternatives. The turbine investment cost calculations were also double-checked and confirmed by our design consultant.

Although the apparent negative-cost paradox has not been fully explained, discussions with our consultants led to the speculation that the fuel cost factor that we were using would not likely have been used in making a design decision relating to thermal efficiency. Rather, a much higher value, reflecting anticipated "escalation" in unit fuel costs would be employed. As a result of these discussions, and also because it promised to shed additional information on other relevant issues, we performed a second general

analysis on the basis of a fuel cost of 33 cents per million Btu—a value about 80% higher than the original 18 cent fuel price and one more nearly reflecting current conditions. All other input parameters and assumptions of the study were left unchanged.

Results for High Fuel Cost Factor

The direct consequences of changing the fuel price from 18 cents to 33 cents/million Btu's are summarized in Table 10, and we need comment only briefly on the implications of this parametric change on absolute values and relative rankings of the resulting steam-cycle cost increments.

TABLE 10
COMPARISON OF STEAM CYCLE INCREMENTAL ANNUAL COSTS
CALCULATED AT ALTERNATIVE FUEL PRICES
OF 18¢ AND 33¢/MILLION Btu (\$10³)¹

Steam Cycle Cases	Capital Cost	Fuel Cost @ 18¢	Fuel Cost @ 33¢	Total Cost @ 18¢	Total Cost @ 33¢	Change in Total Cost
	(1)	(2)	(3)	(1)+(2)	(1)+(3)	(3)-(2)
Low Pressure						
1	Base	Base	Base	Base	Base	Base
2	124	- 86	-158	38	- 34	- 72
3	- 2	- 10	- 18	-12	- 29	- 8
4	124	-110	-202	14	- 78	- 92
Intermediate Pressure						
5	- 71	120	220	49	149	100
6	53	30	55	83	108	25
7	- 74	90	165	16	91	75
8	52	6	0	52	52	0
High Pressure						
9	-201	210	385	9	184	175
10	- 77	112	205	35	128	93
11	-203	190	348	-13	145	158
12	- 77	90	165	13	88	75

¹ For additional costing assumptions see notes to Tables 1 and 2.

One should expect, a priori, that higher fuel costs should lead to the more efficient (lower heat rate) designs being systematically favored; and this is precisely the result, as the increased magnitude of the fuel cost influence now comes to dominate the total steam cycle cost values. Thus, the three low pressure alternatives to the base case, all of which have somewhat better heat rates, now all show cost savings. Conversely, the medium-pressure Cases 5-7 show moderate to substantial increases in incremental costs; and the high pressure Cases 9-12 are now all among the highest cost cases. Indeed, there is a virtually complete reshuffling of relative SIAC rankings among the twelve cases, which now conforms to their relative heat rate rankings. Not only are the relative rankings altered; more importantly for the final outcome, the absolute values are considerably increased, thus giving steam cycle modification costs much more weight in the outcome of the combined steam cycle and cooling system cost synthesis presented in Table 11.

Comparison of Table 11 with the original integrated total cost matrix of Table 9 indicates the extensive changes in final results produced by changing the fuel price parameter from 18¢ to 33¢/million Btu. Although the pattern of individual cooling system types (once-through, helper, etc.) associated with particular effluent standards remains very much the same, the least-cost steam cycle alternatives are almost completely revised. For the higher flowrate standards of 250,000 gpm and above that can be achieved by once-through or helper schemes, the two highest efficiency steam cycle designs—the low pressure Cases 2 and 4—now completely replace the high pressure, low efficiency Case 11 which dominated the previous results. For $L^* < 250$ gpm, the high-pressure Cases 10 and 12 are now the preferred choices in place of their somewhat less efficient counterpart Cases 9 and 11. Although Cases 10 and 12 show substantially higher costs on the steam cycle side than do their lower pressure counterparts, it is evident that the cooling system cost advantages of these high pressure designs with respect to the low flow helper and recycle schemes are still sufficient to more than offset their fuel cost disadvantage.

It may be noted that under the higher fuel cost assumption the total incremental cost results for the least cost systems have, with but one exception, increased by about \$50,000 to \$100,000 compared to the earlier results.

TABLE 11

INTEGRATED MATRIX OF TOTAL INCREMENTAL ANNUAL COSTS FOR LEAST
 COST COMBINED SYSTEMS RELATIVE TO EFFLUENT STANDARDS
 547 MW COAL-FIRED UNIT, CASE NOS. 1-12
 RESULTS FOR HIGH FUEL PRICE ASSUMPTION (\$10³)

Effluent Temperature Standards T* (°F)	Effluent Flowrate Standards, L* (10 ³ gpm)					
	550	450	350	250	150	50
110°	O (4) -\$41	O (4) -\$41	O (4) -\$41	O (2) -\$28	O (12) \$8	R (12) \$229
106°	O (4) -\$41	O (4) -\$41	O (4) -\$41	O (2) -\$28	O (12) \$8	R (12) \$229
102°	O (4) -\$41	O (4) -\$41	O (4) -\$41	O (2) -\$28	H (12) \$54	R (12) \$229
98°	O (4) -\$41	O (4) -\$41	O (4) -\$41	O (2) -\$28	H (12) \$79	R (12) \$229
94°	O (4) -\$41	O (4) -\$41	O (4) -\$41	O (12) \$14	H (12) \$122	R (12) \$241
90°	O (4) -\$41	O (4) -\$41	O (4) -\$41	H (2) \$112	H (12) \$177	R (12) \$266
86°	O (4) -\$22	O (4) -\$4	H (2) \$178	H (2) \$178	H (12) \$254	R (10) \$300
82°	O (4) \$8	H (2) \$313	H (2) \$313	H (2) \$313	R (10) \$365	R (10) \$365
78°	No Feasible Systems Designed for T* ≤ 78°F					

Note, finally, that negative incremental cost values, although reduced in frequency and absolute size, still occur. However, they now occur only for the low back pressure designs with once-through cooling systems that have lower heat rates than our Case 1 base—i.e., those cases that show substantial fuel cost savings at the high fuel price. Under these circumstances the negative values appear entirely consistent with engineering economic principles and current design practice. It is now quite reasonable that among all the alternatives considered in our study, steam cycle Case 4 and its associated least-cost once-through cooling system (as represented by the -\$41,000 value in Table 11) constitute the least-cost overall system, disregarding effluent characteristics.

III. PROPOSED-PLANT DEMONSTRATION—1000 MW NUCLEAR UNIT

Introduction

This section presents a second case study demonstration of our methodology for estimating the costs of meeting thermal effluent standards for proposed steam-electric power plants, this time with reference to a 1000 mw nuclear unit. Nuclear-fueled generating units are of considerable interest owing to their very large typical sizes and (under current technology) relatively high heat rates which combine to produce large quantities of waste heat rejection. In addition, the technical possibilities for steam cycle modification are more restricted than in the fossil-fueled case, thus requiring a somewhat different approach in dealing with potential capacity changes related to steam cycle redesign.

Although the present study examined a more limited set of steam cycle design alternatives and did not explore the questions of reduced capacity or revised utilization as explicit abatement options, in other respects it closely parallels the case study of the 547 mw coal-fired plant in both design and presentation.

The Base Case and Alternative Steam Cycle Designs

This demonstration study is modeled on the basis of a 1000 mw (nominal) prototype nuclear generating unit recently installed on a river site in New York State. Steam is generated by a pressurized water reactor at an approximate rate of 13 million lbs/hr, and delivered to the turbine throttle at a pressure and temperature of 730 psia and 508°F, respectively.

The base case design employs a once-through cooling system utilizing single pressure condensers at a design vacuum of 1.5" Hg abs. In developing steam-cycle modification options, we also consider two alternative design vacuums of 2.5" and 3.5", in order to span the relevant range of possible design vacuums for units of this type and the general north-temperate regional location. Although we consider an examination of alternative dual pressure condenser systems potentially important (as indicated in Section II), we did not have the resources to pursue it here.

In considering possibilities for modifying the steam cycle, the consensus of our Jackson and Moreland power plant design consultants is that for a given condenser vacuum, reactor type, and nominal capacity, the opportunities for improving the station

heat rates are generally nil. This is a result of reactor safety regulations and engineering constraints imposed by current reactor heat transfer technology. Nuclear units thus differ from fossil-fueled units in that it is frequently possible to improve heat rates for fossil units (at least marginally) by increasing capital investments. Consequently, the only steam cycle redesign possibilities considered in this nuclear case study relate to determining the most likely turbogenerator design (from an engineering-economic standpoint) for each of the two alternative design condenser vacuums. The turbogenerators for the alternative vacuums, as developed by our consultant engineers, were derived by modifying the original base case selection in the following ways:

- Increase the steam flow required by the feedwater pump to develop the same pump horsepower as in the original design.
- Reduce the duty of the lowest pressure feedwater heater.
- Reduce the rating of both the turbine and the generator.

Because nuclear steam supply offerings are conventionally available only at standard sizes, and since we wish to maintain at least the nominal capacity of 1000 mw, we are forced to hold the design maximum reactor thermal output and steam throttle flow constant for all alternative designs. Given the same reactor heat output for all designs whereas station net heat rates increase at higher design condenser vacuums, it is impossible to maintain the same level of guaranteed output for all designs. Thus, in contrast with the fossil-fuel case study where it was possible to adjust steam supplies incrementally to hold the guaranteed generating capacity constant in the face of variations in design heat rates, guaranteed capacity for our nuclear unit decreases at the higher design vacuums, and this decrease in capability must be reflected in reckoning the costs of these modifications.

Table 12 summarizes the essential technical features for the base case unit and the two alternative designs, indicating the respective increases in both condenser duty and station heat rate and the decrease in guaranteed capacity as the condenser vacuum is increased. In comparing the base case condenser vacuum of 1.5" Hg abs to the highest vacuum of 3.5", the condenser duty increases by less than 1 percent; however, the heat rate increases by about 5 percent, and the guaranteed capacity decreases by about 5 percent.

TABLE 12
CHARACTERISTICS OF ALTERNATIVE DESIGNS
1000 MW NUCLEAR UNIT*

Design number	Condenser vacuum (in. Hg abs)	Condenser duty (10^6 Btu/hr)	Station heat rate (Btu/kw hr)	Rated capacity (mw)	Capacity difference (mw)	Energy difference [†] (10^6 kw hr/yr)
1	1.5	6991	10,628	1,022	base	base
2	2.5	7008	10,813	1,004	-18	-130
3	3.5	7042	11,189	970	-52	-377

* Unless otherwise indicated, results in this table were developed by H. L. R. Newbold of Jackson and Moreland, Division of United Engineers and Constructors, Inc., Boston, Massachusetts.

[†] Based on an availability of 7884 hrs/yr and average operation at 92 percent of full capacity while available.

Table 13 presents the incremental annual cost estimates for the alternative steam cycle designs. The plant operating assumptions and costing factors used to develop these annualized investment and penalty cost calculations were the same ones employed in the original engineering studies for the prototype unit. Briefly, these calculations are based on a fixed charge rate of 15.8 percent, generating unit availability of 7884 hours per year, average operation of 92 percent of guarantee capacity while available, a capacity penalty demand charge of \$190/kw, and an energy charge of 2.0 mills/kwhr.

TABLE 13
ANNUAL INCREMENTAL COSTS FOR ALTERNATIVE DESIGNS
1000 MW NUCLEAR UNIT
(Costs in $\$10^3$ /yr)

Design number	Investment* (1)	Capacity loss [†] (2)	Energy loss [‡] (3)	Total (1+2+3)
1	base	base	base	base
2	-20	540	260	780
3	-49	1560	754	2265

* Incremental capital investment cost for turbine-generator, as calculated by Jackson and Moreland, times 0.158 (fixed charge rate).

[†] Calculated as capacity difference (Table 12) $\times$ \$190/kw $\times$ 0.158.

[‡] Calculated as energy difference (Table 12) $\times$ 2.0 mills/kwhr.

As in the fossil unit case study, turbine-generator capital investment cost decreases somewhat at the higher condenser vacuums. However, the modest investment cost savings are easily dominated in this case study by the capacity-loss and energy-loss penalty costs, both of which are quite sizable in absolute terms at the higher design vacuums as a result of the capacity reductions. Note that the costs presented here are exclusive of those for the condenser and other components of the circulating water system which are developed in the next stage of the analysis.

Cooling System Design Results

Having described the three alternative steam cycle designs, we now present results from the second stage of the overall procedure in which cooling system design computer runs were performed for each of the three steam cycle alternatives. Recall that all cooling system design variations produced in a given computer run have the same pre-set design-condition values for condenser duty and vacuum (from Table 12), serving to link and maintain consistency between any given steam cycle design and its corresponding cooling system design options. From the array of technically feasible cooling system alternatives produced in each computational run of the cooling system design program, the least-cost cooling system is selected for each of a pre-set schedule of alternative effluent flowrates (L^*) and effluent temperature (T^*) constraints. For these particular case study runs, nine (9) alternative values of L^* ranging from 1,700,000 gpm down to 100,000 gpm, in equal increments of 200,000 gpm, were specified, together with eight (8) alternative values of T^* , from 107°F to 75°F, in 4°F increments. Other specific input values for the 1000 mw nuclear unit case study are listed in Appendix D.

The base-case cooling system design was determined within the computer run for Case 1 as the least costly cooling system among all alternatives produced for the base steam cycle design, irrespective of any thermal effluent quality considerations. The once-through cooling system thus selected was designed for a flowrate of 1,420,000 gpm and a summer design effluent temperature of 84.7°F, a temperature rise of 9.7°. The total annual cost for this "privately optimal" cooling system was approximately \$1.6 million per year, which thus provided the base-case cost estimate against which all other cooling system design variations for all runs in this plant study were compared for purposes of calculating their incremental cooling system cost.

The particular sets of input values for technical design parameters and program operating instructions utilized in the three cooling system design runs for this case

study resulted in the generation of a total of 828 individual cooling system designs for the three runs taken together. Of these, 21 designs were produced for the 1.5" vacuum case, 274 for the 2.5", and 533 for the 3.5" condenser vacuum alternative. As noted in previous sections, it is a general characteristic of our cooling system design program that, other things equal, higher design condenser vacuums (i.e., higher steam condensing temperatures) allow for greater design variations in cooling water temperature rise and flowrate; and this in turn results in larger numbers of alternative designs being produced for each alternative type of cooling system at the higher design vacuums.

Many of the general characteristics of the output resulting from the cooling system design runs closely paralleled those from the fossil fuel case study. To a large extent, this was to be expected, since, with the exception of differences in scale (the 1000 mw nuclear unit having roughly three times the condenser heat load of the 550 mw fossil unit) the two case studies did not differ appreciably in terms of most of their cooling system design parameters. Thus, for example, we would find qualitative cost function relationships for individual cooling system types and relative patterns among types similar to those previously illustrated in Figure 1 and 2, and Table 6. For the particular input values employed, cooling ponds once again turned out to be somewhat less costly, case-for-case, than mechanical draft towers designed for the same conditions. Since the reasons for this were analyzed in some detail for the fossil-plant case, they will not be repeated here.

Absolute total annual costs for cooling systems designed in this case study are of course substantially higher than those of the previous study—on the order of 3.5 to 4 times higher for comparable schemes—due both to the much higher condenser duties and the almost 40 percent higher fixed charge rate specified in the nuclear example. Thus, for example, for the three design runs developed in the present instance, we are dealing with once-through cooling systems costing between \$630 thousand and \$1.8 million per year and least-cost recirculation ponds ranging from \$1.6 to \$4.6 million per year (the high values being for 1.5" vacuum and the low values for the 3.5" vacuum designs).

These large absolute values imply correspondingly large incremental cost differences with respect to the base-case (once-through) cooling system designed for the 1.5" vacuum, which was estimated at \$1.6 million per year. For alternative once-through designs, incremental cooling system costs thus involved rather significant

negative values (i.e., savings) of up to \$780 thousand for the least-costly 2.5" vacuum design and up to \$965 thousand for the least costly 3.5" vacuum design. At the other extreme, the least costly recirculation pond for the 1.5" case had an incremental annual cost of over \$3 million per year over the base case once-through system; however, this decreased substantially to \$580,000 at the 2.5" vacuum and, further to a mere \$14,000 at the 3.5" vacuum.

Values such as these serve to underscore the impressive cooling system cost differences for given cooling system types achievable from differences in design back pressure selection. It may be noted in passing that the largest cost reductions occurred for the first pressure increment from 1.5" to 2.5", and that the second increment to 3.5" had much less absolute impact on the cost of given cooling system types. In other words, it appears as though we are witnessing another example of the general phenomenon of diminishing returns.

It may also be noted that cooling system incremental costs in this nuclear unit case study are of the same order of magnitude as the incremental costs of steam-cycle modification. Since steam cycle incremental costs vary directly with design back pressure while cooling system incremental costs vary inversely with back pressure, we have the basis for an interesting trade-off analysis in selecting the least-cost combinations of steam cycle and cooling systems for meeting alternative possible effluent standards.

Integrated Results for the 1000 MW Nuclear Unit

The overall results for the 1000 mw nuclear unit are presented in Table 14 which integrates the results of combining steam cycle incremental costs with the respective cooling system incremental cost matrices for all three design cases considered. As such, Table 14 is comparable in definition and format to Table 11 for the fossil-fueled unit.

It is evident that the base-case steam cycle designed for a 1.5" condenser vacuum remains the least cost selection only for the range of effluent standards defined by $L^* > 1,300,000$ gpm and $T^* > 79^\circ\text{F}$, in spite of the fact that, by definition, Case 1 involves zero steam cycle penalty costs. A total incremental annual cost matrix for the Case 1 run considered separately would indicate an extremely limited number of feasible cooling system alternatives, with the only designs capable of achieving effluent standards of $L^* \leq 1100$ gpm being of the high cost recirculation type. This

TABLE 14
INTEGRATED MATRIX OF TOTAL INCREMENTAL ANNUAL COSTS
FOR LEAST COST COMBINED SYSTEMS RELATIVE TO EFFLUENT
STANDARDS 1000 MW NUCLEAR UNIT, RUN NOS. 1-3 (\$10³)

Effluent Temperature Standards T* (°F)	Effluent Flowrate Standards, L* (10 ³) gpm								
	1700	1500	1300	1100	900	700	500	300	100
107°	0 (1) \$0.0	0 (1) \$0.0	0 (2) \$1.0	0 (2) \$1.0	0 (2) \$9.0	0 (2) \$65	0 (2) \$471	R (2) \$1361	R (2) \$1361
103°	0 (1) \$0.0	0 (1) \$0.0	0 (2) \$1.0	0 (2) \$1.0	0 (2) \$9.0	0 (2) \$65	H (2) \$512	R (2) \$1361	R (2) \$1361
99°	0 (1) \$0.0	0 (1) \$0.0	0 (2) \$1.0	0 (2) \$1.0	0 (2) \$9.0	0 (2) \$65	H (2) \$599	R (2) \$1361	R (2) \$1361
95°	0 (1) \$0.0	0 (1) \$0.0	0 (2) \$1.0	0 (2) \$1.0	0 (2) \$9.0	H (2) \$275	H (2) \$791	R (2) \$1361	R (2) \$1361
91°	0 (1) \$0.0	0 (1) \$0.0	0 (2) \$1.0	H (2) \$1.0	H (2) \$313	H (2) \$459	H (2) \$1009	R (2) \$1398	R (2) \$1398
87°	0 (1) \$0.0	0 (1) \$0.0	B (2) \$17	H (2) \$441	H (2) \$606	H (2) \$795	H (2) \$1301	R (2) \$1557	R (2) \$1557
83°	H (1) \$1182	H (1) \$1182	H (2) \$1204	H (2) \$1204	H (2) \$1204	H (2) \$1279	H (2) \$1750	R (2) \$1861	R (2) \$1861
79°	H (2) \$2153	H (2) \$2153	H (2) \$2153	H (2) \$2153	H (2) \$2153	H (2) \$2153	R (2) \$2608	R (2) \$2608	R (2) \$2608
75°	No Feasible Systems Designed for T* ≤ 75°								

is due to the very restrictive maximum temperature-rise constraint imposed on once-through and helper designs for Case 1 by the very low 1.5" design back pressure condition (implying a steam condensing temperature of only 91.7°F), and a consequently high limit on the minimum feasible circulating water flowrate.

On the other hand, Case 2 with a design vacuum of 2.5" (steam condensing temperature of 108.7°F) was capable of producing a number of alternative once-through and helper schemes with effluent flows satisfying L*'s down to 500,000 gpm. Thus, not only were Case 2's cooling systems generally less costly for comparable types and effluent flows than those of Case 1, but they also provided considerable scope for substitution of lower cost once-through and helper schemes in place of the more costly recirculation

types over the intermediate flowrate standards as contrasted with Case 1. At the lowest flowrate standards of $L^* \leq 300,000$ gpm, where recirculation schemes were the only technically feasible designs even for Case 2, it was still possible for Case 2 to show substantially lower total incremental costs than Case 1 in spite of Case 2's net steam cycle penalty costs of \$780,000 per year, owing to the fact that Case 2's best recirculation designs were about \$1.8 million per year less costly on the cooling system side of the accounts than those of Case 1. A similar though less dramatic cooling system cost advantage explains the choice of a Case 2 helper scheme for achieving the low temperature standard of 79°F, even at the higher flowrates where a Case 1 helper design was technically competitive.

The very high capacity and energy cost penalties for Case 3, giving this 3.5" vacuum design a net incremental cost on the steam cycle side of the accounts of almost \$2.3 million per year, explains why Case 3 did not place in the final combined results shown in Table 14. Thus, even though Case 3 cooling systems showed a consistent cost advantage over those of Case 2 (ranging up to \$500,000 per year in some instances) this was not enough to outweigh the additional steam cycle penalty costs of almost \$1.5 million. However, it should be noted that in spite of its almost \$2.3 million cycle cost increment over Case 1, Case 3 would still have had lower total incremental costs than the base-case alternatives in achieving all flowrate standards of $L^* \leq 1,100,000$ gpm.

In summary, considering those effluent standards that cannot be met by the base-case steam cycle and cooling system—i.e., flowrates less than 1,420,000 gpm and/or design summer effluent temperatures less than 84.7°F—our analysis for the 1000 mw nuclear unit indicates that significant control of either of these effluent characteristics is quite costly in absolute terms, but that for most alternative effluent standards there are very significant abatement cost advantages to be gained from considering alternative combinations of steam cycle parameters and cooling system types. Indeed, for nuclear units at the 1000 mw scale, selection of either a 1.5" or a 3.5" instead of a 2.5" back pressure design condition can mean additional abatement costs of well over a million dollars per year for achieving certain standards. In the present case study, penalty costs for capacity and energy differences had a decisive influence on total abatement cost levels and, especially, on choices between the higher-vacuum Cases 2 and 3.

Addendum to the 1000 mw Nuclear Unit Case Study

After the preceding analysis for the 1000 mw nuclear unit had been completed, it was determined that an incorrect value for the design condenser vacuum of the prototype plant had been utilized as the basis for developing our engineering design and incremental cost calculations. In fact, the correct condenser vacuum for the prototype was found to have been 2.0" Hg rather than the 1.5" erroneously employed as the base case design condition in our case study. Unfortunately, this discovery occurred too late to allow for the necessary recomputations for inclusion in this report.

This does not mean that the results developed in the study are necessarily invalid or irrelevant. (Indeed, it was precisely the results from our cooling system design analysis in developing a "base-case" cooling system that caused us to question the validity of the 1.5" vacuum as the correct value for the prototype unit). However, this does mean that our results as presented here should be interpreted with caution.

Essentially, our "base-case" at a 1.5" vacuum in fact represents a rather atypical nuclear unit designed for close to maximum practical thermal efficiency, given its very low steam condensing temperature of 91.7°F. Our computer program calculated that an optimal once-through cooling system for this unit would utilize an extremely large circulating water flowrate in excess of 1,400,000 gpm and operate at an uncommonly low design temperature-rise across the condenser of less than 10°F.

In contrast, a base case designed for a 2.0" condenser vacuum would, realistically, represent a rather more typical example of current design practice based on economic considerations. Rough calculation suggests that our cooling system design program would produce an optimal once-through cooling system for the 2.0" vacuum design having a design temperature-rise closer to 15°F, with a flowrate closer to 900,000 gpm, and that it would cost on the order of several hundred thousand dollars per year less than the previously estimated base-case cooling system. Were we to thus redefine our base case cooling system (the new base being less costly), the whole structure of incremental cooling system costs originally calculated for our Case 1 (1.5" vacuum), Case 2 (2.5" vacuum) and Case 3 (3.5" vacuum) cooling systems would all be revised upwards by the amount of the decrease in the base cost. Thus, selection of an incorrect base case in this study had the general effect of understating the incremental costs and overstating incremental savings (for those systems that were less costly than the original base) on the cooling system side of the accounts. However, this 2.0" back pressure generating

unit would operate at slightly higher heat rate and slightly lower rated capacity than the 1.5" design. That is, based on the same prototype thermal reactor steam-supply source, its design characteristics would probably lie about halfway between those of our Case 1 and Case 2 shown in Table 12; and its annual steam cycle cost differentials would likewise lie between those for Cases 1 and 2 in Table 13. Had the 2.0" vacuum design alternative then been utilized as our base case for purposes of calculating cost differentials, the total incremental penalty costs for Cases 2 and 3 as presented in Table 13 would then have been some \$300,000 to \$400,000 per year less than the values utilized in our subsequent analysis. (Case 1 would now show negative penalty costs [i.e., net benefits] of this amount on the steam cycle side of its accounts.) This would have had the effect of reducing the total combined incremental abatement costs of all three original cases by this amount.

In summary, redefining our base case to correspond to the prototype's actual design vacuum of 2.0", rather than the 1.5" value, has the effect of subtracting a constant amount (estimated at \$300,000 to \$400,000 per year) from the steam cycle penalties and at the same time adding a constant amount (of the same order of magnitude) to the incremental cooling system costs for all alternative designs. Although not precisely known, the net effect on the cost structure of our results as presented in Table 14 is thus likely to be rather less than one might have supposed. The general pattern of cooling system types for achieving particular standards would be altered very little, if at all; although one would expect that a new "Case 1A" representing designs based on the 2.0" vacuum would displace those for the present Case 1 at the high flowrate standards and possibly also some of the Case 2 designs for the intermediate flowrate standards.

In conclusion, this "addendum" serves to emphasize the basic problem one faces in any attempt to evaluate the cost of controlling a waste discharge for a "proposed" plant, namely, that of first being able to estimate correctly what the design conditions and production costs would have been in the absence of the control program.

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APPENDIX A

TECHNICAL SPECIFICATIONS FOR THE CEM COOLING SYSTEM DESIGN PROCEDURE FOR PROPOSED PLANTS

A.1 Introduction

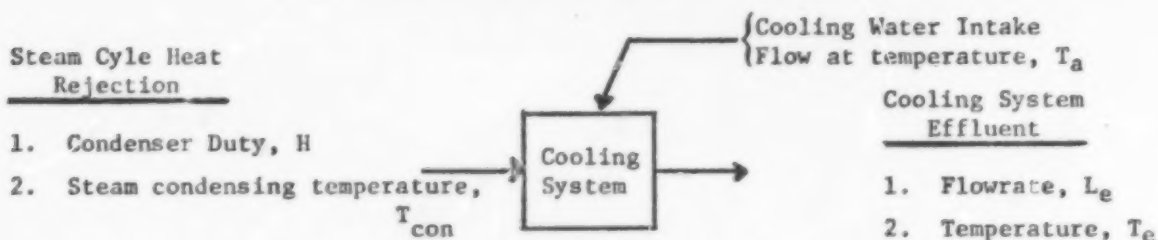
In Section I a sequential procedure for deriving and evaluating alternative waste heat abatement programs was presented. The overall objective of this sequential procedure is the determination of that combination of abatement options that minimizes abatement costs while meeting constraints on power output, and satisfying a specified thermal quality standard. The abatement options considered include variations in plant design capacity, steam cycle design (i.e., station heat rate), plant utilization schedule, and variations in cooling system design.

This appendix considers the procedures employed in examining the last of the listed abatement options, namely, variations in cooling system design. Since the approach discussed in this appendix was developed exclusively as a component of the sequential procedure described in Section I, it differs somewhat from conventional approaches to cooling system design which generally size cooling system components to minimize the contribution of the cooling system to the total power production cost, but without systematic treatment of thermal quality standards. Consequently, the approach discussed herein cannot be viewed as a complete procedure in and of itself; rather, it must be viewed as part of the broader analysis introduced in Section I. In this connection, observe that the analysis developed in this appendix is not meant to be performed a single time for a given plant; it is to be performed a number of times, once for each alternative combination of plant design capacity, steam cycle design, and plant utilization schedule.

Recall from Section I that the analysis of alternative cooling system designs takes, as given, the plant design capacity, steam cycle design and plant utilization schedule. The final "stage" in the sequential procedure consists of determining cooling system designs that transfer H Btu/hour at the condenser at a specified design steam condensing temperature, T_{con} , and simultaneously meet the pre-imposed thermal quality standards in the form of upper limits on the cooling system effluent's temperature, T_e , and flow rate, L_e . From among such cooling system designs it is necessary to select the one having the lowest costs for the given H and T_{con} . This cooling system design analysis is repeated, in its entirety, for each combination of plant design capacity, steam cycle design, and plant utilization schedule being considered. (Each such combination will involve different values of H and, in many cases, T_{con} .) The final step in the procedure for selecting

the least costly combination of waste heat abatement options integrates these analyses of alternative cooling system designs with the cost characteristics relevant to the system cycle and other options.

The sketch below serves to put the cooling system in perspective with respect to both the steam cycle and the water body receiving the heat effluent. Inputs to the cooling system include the heat rejected from the steam cycle at the condenser, and the cooling water intake flow. The outputs of concern here include the cooling system effluent flowrate, L_e , and temperature, T_e . The physical components of the cooling system itself include pipes (and/or tunnels or canals) used for intake and discharge, pumps, water treatment facilities (if any), a condenser, and possibly a heat rejection component (e.g., a mechanical or natural draft cooling tower, or a cooling pond).



Our definition of the cooling system specifically excludes components and considerations relevant to the ultimate disposal of the effluent in the receiving water. Thus, for example, the cooling system as we have defined it does not include components such as skimmer walls, underwater dams and diffusers. These components, far from being unimportant, are intimately related to features of the receiving water that are very much "location specific." Considerations relevant to selecting such components, as well as the all important decision on effluent outfall location, would use the overall analysis of this report as an input. Thus, while we occasionally use the term "optimum" as a matter of convenience, we fully recognize that, with respect to such larger issues, we are performing a "suboptimization study."

This appendix describes the assumptions, formulas and procedure embodied in a computer program developed to produce the cooling system designs needed for the waste heat abatement analysis described in Section I*. Essentially this

* The computer program (prepared in FORTRAN and run on an IBM 360/65), together with sample input data and other documentations, has been submitted to the National Thermal Pollution Research Program, Pacific Northwest Water Laboratory, 200 S. 35th Street, Corvallis, Oregon 97330.

procedure consists of the systematic determination of an array of alternative designs for each of three broadly defined cooling system flow configurations, namely, once-through, recirculation, and helper. Each such individual design yields different values of T_e and L_e . Once all designs are produced a simple sorting routine reveals that subset of designs having effluents meeting given alternative thermal quality standards; and, within that subset, the minimum cost design meeting each such standard is selected.

The computations involved in the cooling system design procedure and the organization of the major sections of the computer program are presented in the remaining portions of this appendix. First we define the parameters and functions that are taken as "given" in the context of the design procedures, i.e., the inputs to the computer program. Then, for each of the major design types--once-through, recirculating, and "helper"--we present the following: a diagram showing the pipe layout, a flow diagram showing the essential steps in the computation, and a discussion of the main formulas used. Following this, we describe how the information generated for different design types can be synthesized in the deduction of minimum cost designs. Finally, we provide details regarding the assumptions used in the design and costing of the pipes, pumps, condenser and heat rejection devices, and in the determination of operation and maintenance costs.

A.2 Listing of Input Variables Employed

For reference purposes, the classified listing below provides a glossary of the technical input variables and economic cost parameters utilized in the development of alternative cooling system designs for proposed power plants. Two forms of symbolic notation have been employed in our work, and both are presented here. The first and shorter form is that used throughout the present appendix for convenience in describing procedures and expressing mathematical arguments. These are the symbols shown at the left. The second is the computer language form, for which the symbolic variable names have been presented in brackets to the right of the written variable names, as a cross-referencing aid to those who may actually become involved in working with the program.*

Cooling System Design Conditions Related to Steam Cycle Design

H	= condenser duty (Btu/hr).	[H]
T _{con}	= steam condensing temperature (°F).	[TCON]

Site-Specific Climatologic and Hydrologic Conditions

L	= maximum feasible design intake flowrate (gal/min).	[BARL]
T _a	= design ambient temperature of cooling water source (°F).	[TA]
T _w	= design wet bulb temperature (°F).	[TWB]
T _d	= design dew point temperature (°F).	[TDEW]
R _h	= design relative humidity (%).	[RH]
E	= initial estimate of equilibrium water temperature (°F).	[TEHAT]
S	= design daily solar radiation (Btu/ft ² /day).	[SORAD]
W	= design windspeed (miles/hr).	[WSPEED]

Condenser

C ₁	= empirical constant in heat transfer derivation.	[CFUDG]
V	= velocity of cooling water (ft/sec).	[VEL]
α _c , β _c , γ _c	= coefficients in condenser capital cost function.	[ACON] [BCON] [CCON]
d _c	= inside diameter of condenser tubes (inches).	[DCON]
d' _c	= outside diameter of condenser tubes (inches).	[DOCON]
D _c	= length of a single condenser tube in optimal design (ft).	[TUBEL]

* A complete input listing, including both technical input and program control inputs, with card format has been provided separately to the Water Quality Office, EPA, Pacific Northwest Water Laboratory.

K_1 = an integer giving the number of passes in condensers [FPASS]
 R_{min} = minimum temperature increase through condenser ($^{\circ}F$) [RISMMO]

Mechanical Draft Tower

α_m = unit cost in terms of \$ per "tower unit." [TUDOL]
 c_m, r_m, a_m, w_m = coefficients in function for tower rating factor. [AM(I)]
 R_{min} = minimum temperature decrease (range) across tower or (or pond)*. [RISMM]
 A_{min} = minimum "approach" to wet bulb ($^{\circ}F$) [APMM]

Natural Draft Tower

c_n, r_n, a_n, w_n, h_n = coefficients in capital cost function. [CN,RN,AN,WN,HN]

Cooling Pond

α_L = unit cost for land, site preparation, and construction of dikes or dam (\$/surface acre). [CTLD]

Pipes

$D_1, \dots, D_9$ = pipe lengths (ft)--keyed to system layout diagrams. [PLEN(I)]
 C_{HW} = Hazen-Williams smoothness coefficient. [CHW]
 α_i = multiplicative constant in capital cost function. [APIPE]
 β_i = exponential constant in capital cost function. [BPIPE]
 α_f = special constant to account for variable foundation conditions. [PFUDG]

Pumps

$h_1, \dots, h_4$ = elevation heads (ft)--keyed to system types**. [HEAD(I)]
 α_p = multiplicative constant in capital cost function. [APUMP]
 β_p = exponential constant in capital cost function. [BPUMP]
 η_p = pump design efficiency (%/100). [PEFC]

Water Supply and Treatment

α_w = unit cost of purchased water (c/1000 gal). [CTWTS]
 α_{ti} = unit cost of water treatment, using heat rejector i (c/1000 gal). [TRTCT(I)]
 w_i = wind loss as a fraction of flow through heat rejector ($i, i=1, \dots, 3$). [WIND(I)]

* Also applicable to Natural Draft Towers and Ponds.

** Actually, h_i is the elevation head associated with heat rejector i , where i is an index variable; values of i from 1 to 4 correspond to mechanical tower, natural tower, pond, no heat rejector (i.e., once through). The latter is simply the head lost in lifting water to the condenser. Note that while we restricted ourselves to only three heat rejector types, the approach can be extended to accommodate a larger number of types.

- k_{h1} = factor (≤ 1) to allow for possible reduction in water treatment cost when heat rejector 1 is used in a helper system. [TFUDG(I)]
- N = number of cycles of concentration in recirculating water system. [XNC(I)]

Other Cost Related Parameters

- F_1 = O&M cost correction factor. [XLF]
- F_2 = fixed charge rate (applicable to all capital cost items). [XFCR]
- γ_1 = penalty cost for energy for pumps and fans (\$/kwhr). [PCTPF]
- γ_2 = penalty cost for capability loss (\$/kw/yr). [YRPEN]
- α_{mi} = annual maintenance cost factor--as a fraction of total capital cost for system involving heat rejector $i; i=1, \dots, 4$. [CTMT(I)]

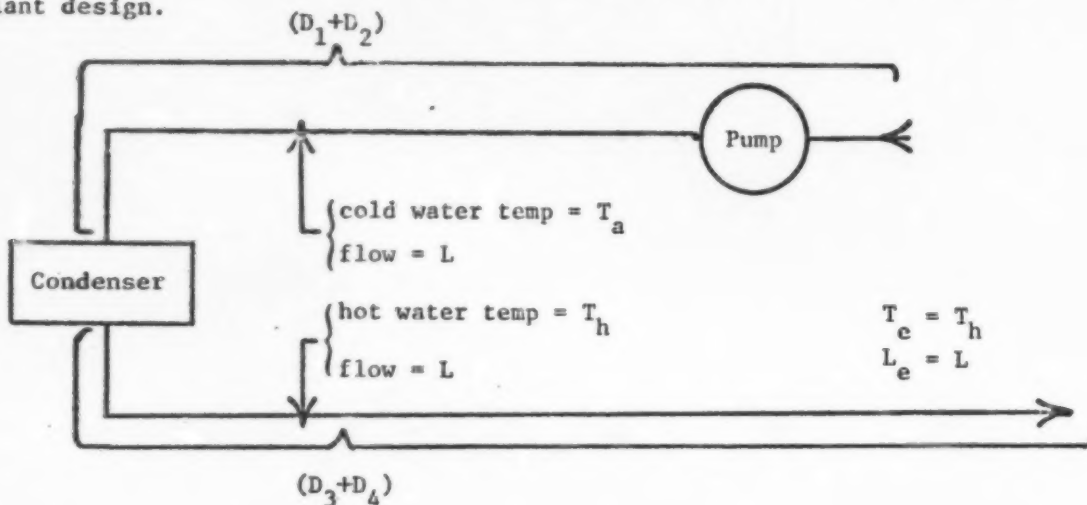
Program Instruction Variables

With respect to the computer program, there are a number of inputs that serve to guide the selection of the schemes to be examined (any one or more of the 8 types included) and the types of output desired to be printed. In addition, other inputs control the total number of designs of each cooling system type to be examined (i.e., the level of detail in terms of incremental changes in flow rates, temperature rises, etc., to be produced for each type), as well as the grid size for abatement-cost output displays. These various program control variables will be introduced, as necessary, in the following discussion.

A.3 Once Through Systems - With and Without Blending

(1) Pipe Layout and Other System Components

The once-through system-without blending consists of intake and discharge pipes, circulating water pumps located near the point of intake, and a condenser.* A diagram of the system is given below. Note that, for reasons of convenience in considering other types of cooling systems, the pipe lengths are given as (D_1+D_2) and (D_3+D_4) . Values of (D_1+D_2) and D_3+D_4 are simply the total length of intake and discharge pipe, respectively, for once-through systems. These values are given by engineering studies for the privately optimal plant design.



For given condenser duties and steam condensing temperatures, the development of alternative once-through designs essentially involves the manipulation of circulating water flows and condenser characteristics to yield various values of effluent temperature and flowrate.

For once-through designs, L , the flowrate of water passing through the condenser, equals L_e , the effluent flowrate; and T_h , the temperature of water leaving the condenser, equals T_e , the effluent temperature.

For every once-through design (characterized by a given value of T_h and L), it is also possible to develop additional alternative designs corresponding to different levels of blending; i.e., the mixing of water passing through the condenser with intake water that bypasses the condenser. Suppose that the maximum possible inflow, based on hydrologic considerations is $\bar{L}$. So long as L , is less than $\bar{L}$, it is possible to mix a portion (or all) of the quantity $(\bar{L}-L)$ at temperature T_a , with the flow L at temperature T_h . Assuming that

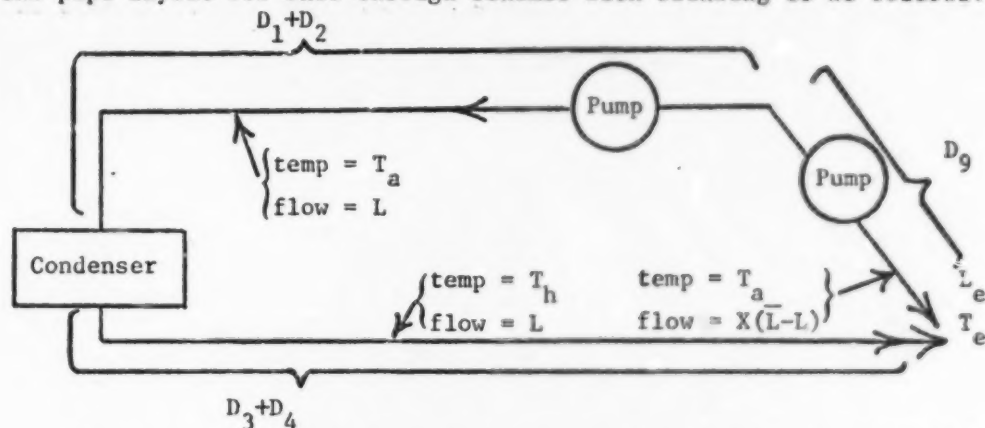
* Water treatment facilities and intake structures are not sized in our procedure; however, related costs are accounted for in the operation and maintenance cost computations. This applies for recirculation systems and helper systems as well. (See subsection A.7.5 below.)

X (where $0 \leq X \leq 1$) is the fraction to be blended, the resulting effluent flow-rate and temperature are determined as follows:

$$L_e = L + X (\bar{L} - L), \text{ and}$$

$$T_e = \frac{L T_h + X (\bar{L} - L) T_a}{L_e} *$$

The pipe layout for once-through schemes with blending is as follows:



Computations needed, beyond those associated with once-through designs, involve the sizing and costing for the additional pump and pipe. For purposes of program execution the actual number of blending schemes associated with a given once-through design is a question for determination by the analyst. Once decided, it is introduced as an instructional input variable to the computer program.

(ii) Computational Procedure for Once-Through System Designs

The flow chart for the computational procedure is given as Figure A.3.1. Below we describe the individual steps in the sequence of computations.

1. Set T_h , the temperature of water leaving the condenser.

Before establishing a value for T_h it is necessary to determine the maximum and minimum values that it can attain; we note these as $T_{h \max}$ and $T_{h \min}$, respectively. Formulas used for computing these values are

$$T_{h \max} = T_{\text{con}} - 5, \text{ and}$$

$$T_{h \min} = T_a + R_{\min}$$

The basis for the first equation is that, as a matter of engineering practice, condensers are designed with at least a 5°F difference between the temperature of the condensing steam [Ref. 1, p.10]. The

* Note that this is simply the weighted average temperature of the two flows.

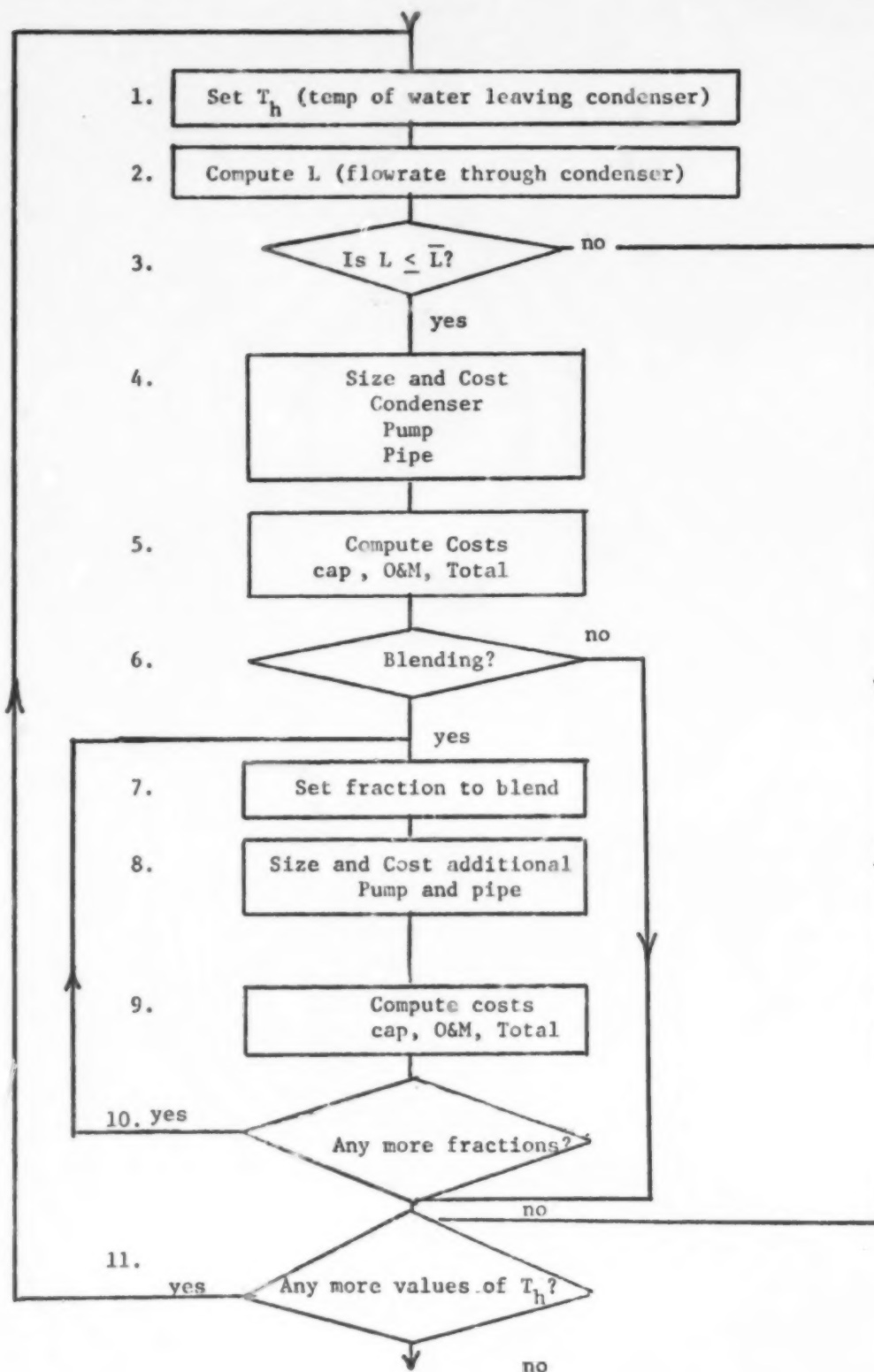


Fig. A.1 Flow Chart for Once-Through System Design Computations

basis for the second equation is that T_h must clearly be greater than T_a if any steam is to be condensed. The equation implies that T_h must be at least $R_{\min}$ (i.e., minimum rise) greater, where $R_{\min}$ is an input parameter selected on the basis of judgment.

In the computer program, developed values of T_h were set recursively.* For the first design $T_h = T_{h \max}$, for the second design $T_h = T_{h \max} - \Delta T_1$, etc. This was organized for digital computation by defining n_1 as the greatest integer in the quantity $(T_{h \max} - T_{h \min})/\Delta T$. Values of T_h were then selected recursively using

$$T_h = T_{h \max} - (j-1) \Delta T_1$$

where j is an index variable on the number of values of T_h examined, and ranges from 1 to (n_1+1) .

2. Compute L

For a given value of T_h , the circulating water flowrate is found using

$$L = \frac{H}{500 (T_h - T_a)},$$

where L is in gal/min, H is the given condenser duty in Btu/hour and all temperatures are in degrees Fahrenheit. This is a basic equation from elementary thermodynamics, and the constant 500 incorporates the specific heat of water as well as the factors required to yield consistent units [Ref. 16 p.32]. For the range of temperatures relevant to these computations, the specific heat of water may be assumed constant.

3. Is $L \leq \bar{L}$?

This comparison is necessary to guarantee that the computed flowrate is no greater than the flow available for diversion. If $L \leq \bar{L}$ establish the identification number associated with this design, and continue; otherwise go to Step 11.

*This is one of many possible ways that alternative values of T_h might be selected as an issue of logic. We chose it as an operating method on grounds of convenience.

4. Size and cost condenser, pump and pipe

The formulas and assumptions used in obtaining estimates of sizes and capital costs for the condenser, pipe and pump are discussed in subsections A.7.1, A.7.3, and A.7.4, below. These subsections contain the assumptions and formulas used in the computer program subroutines for sizing condensers, pipes, and pumps, respectively.

5. Compute costs

The capital costs for the system are taken as the sum of the capital costs for the condenser, pump and pipes. The annual operation and maintenance (O & M) costs are computed for water supply if purchased, and treatment, auxiliary power required for pumps, and maintenance. The assumptions used in these computations are discussed in subsection A.7.5, below. The total annual cost is found, following the practice used in the power industry as the computed annual O & M cost plus the product of a given fixed charge rate times the computed capital cost.

At this point a completed design has been produced and associated costs have been estimated for a particular once-through cooling system without blending.

6. Blending?

In our computer program the decision to consider blending is established by an input parameter. If blending is to be considered, then an identification number is established for this design, the procedure continues with Step 7; otherwise we proceed to Step 11.

7. Set fraction to blend

This is done recursively by defining an input parameter n_b as the number of alternative blending schemes associated with a given once-through design. In addition, X is defined as the fraction of $(\bar{L}-L)$ that is blended; it is computed using

$$X = \frac{k}{n_b}$$

where k is an index variable on the number of values of X considered, and ranges from 1 to n_b . Thus, the first once-through with blending design blends $\frac{1}{n_b} (\bar{L}-L)$, the second blends $\frac{2}{n_b} (\bar{L}-L)$, etc.

Once the value of X is established, the blend flow is determined as $X(\bar{L}-L)$; values of L_e and T_e are found using

$$L_e = L + X(\bar{L}-L), \text{ and}$$

$$T_e = \frac{L T_h + X(\bar{L}-L) T_a}{L_e}$$

8. Size and cost additional pump and pipe

Subsections A.7.4 and A.7.3 contain the assumptions and formulas employed.

9. Compute costs

The capital costs associated with the design are those for the associated once-through without blending plus the capital costs for the additional pump and pipe. The annual O & M costs are those for the associated once-through without blending plus additional costs for water supply, auxiliary power for pumping, and maintenance. The total annual cost is the annual O & M cost plus the product of the fixed charge rate times the capital cost.

10. Any more fractions?

A test is made to see if all blending fractions have been considered. If they have (i.e., $X = 1$) then the procedure continues with Step 11; otherwise we return to Step 7, get an increased value of X , and perform Steps 8-10 again.

If at this point $X = 1$, we have a total of n_6 blending designs associated with the once-through system produced by Steps 1 through 5.

11. Any more values of T_h ?

A test is performed to see if the current value of T_h is the lowest of those to be considered (i.e., to see if n_1+1 values of T_h have been selected?). If the answer is, "Yes," then the design procedure for once-through with or without blending is completed; otherwise we return to Step 1, get a smaller value of T_h , and continue with Steps 2, 3, etc.

The step-by-step procedure described above yields a total of $n_b(n_1+1)$ designs, where n_b is an input parameter indicating the number of blending schemes, and (n_1+1) is the total number of once-through designs not involving blending. Recall that the quantity n_1 is computed as

$$n_1 = \left\lceil \frac{T_{h \max} - T_{h \min}}{\Delta T_1} \right\rceil,$$

where the heavy square brackets, a standard mathematical notation, indicate that n_1 is the greatest integer of the value within the brackets. Recall, also that

$$T_{h \max} = T_{\text{con}} - 5, \text{ and}$$

$$T_{h \min} = T_a + R_{\min}.$$

Substituting these values into the equation for n_1 gives

$$n_1 = \left\lceil \frac{T_{\text{con}} - 5 - T_a - R_{\min}}{\Delta T_1} \right\rceil.$$

We introduce an example to indicate clearly just how many designs are developed. Suppose the input values were specified as follows:

$T_{\text{con}} = 108^\circ\text{F}$; $T_a = 77^\circ\text{F}$; $R_{\min} = 3^\circ\text{F}$, $\Delta T_1 = 5^\circ\text{F}$, and $n_b = 3$. The value of $T_{h \max}$ is

$$T_{h \max} = 108 - 5 = 103^\circ\text{F};$$

and the value of n_1 is

$$n_1 = \left\lceil \frac{108 - 5 - 77 - 3}{5} \right\rceil = \left\lceil \frac{23}{5} \right\rceil = 4.$$

There would be a total of five once-through without blending designs, corresponding to values of T_h equal to 103, 98, 93, 88 and 83°F . For each once-through without blending design corresponding to a value of T_h , there would be schemes; these correspond to blending $1/3$ of $(\bar{L}-L)$, $2/3$ of $(\bar{L}-L)$ and all of $(\bar{L}-L)$. In total, there would be 15 designs.

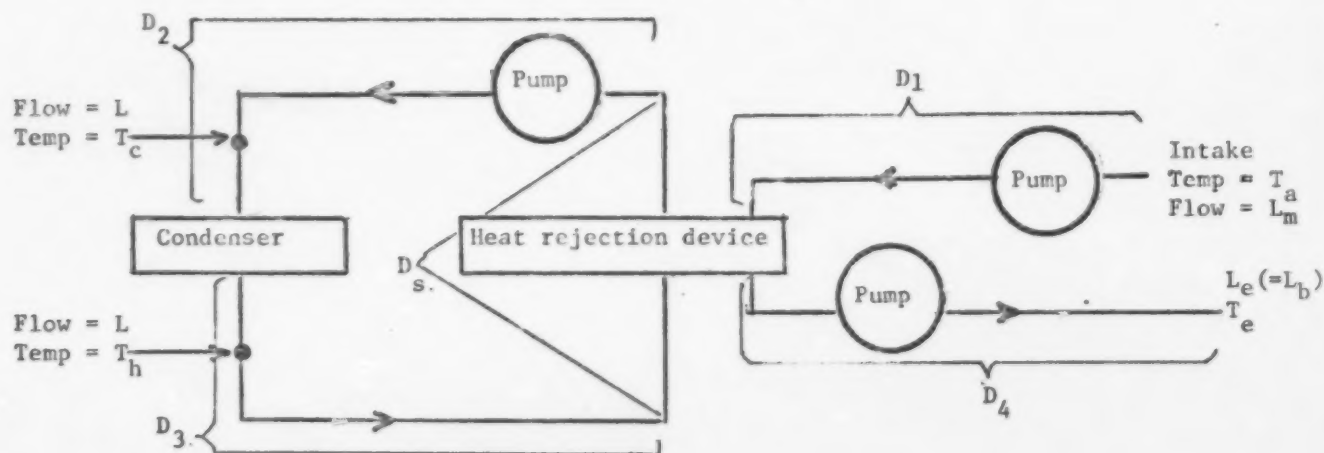
A.4 Recirculation Systems

Pipe Layout and Other System Components

Cooling systems involving recirculation are defined herein to consist of the following elements:

- An intake pipe and pump for makeup water.
- A discharge pipe and pump for blowdown.*
- Pipes and pump for recirculating water.
- A condenser.
- A heat rejection device (i.e., cooling tower or pond).

The diagram below shows the pipe layout used for all recirculation system designs; the variables D_i refer to pipe lengths and are input quantities in our computer program formulation.**



A particular recirculating system design may be described in a number of ways. For the level of detail employed in the analysis described below, such a description could include the following: effluent temperature and flowrate, make-up water flowrate, condenser area, pipe diameters, pump horsepower, component and total cost estimates and, where applicable, cooling pond areas and the rate of heat rejected by cooling towers.

* We also deal with cases where blowdown can be accomplished by gravity flow, and consequently a blowdown pump is not required.

** These values are selected using judgment guided by the spatial layout of the privately optimal design.

Using the procedure described herein, all of the above mentioned characteristics can be deduced once the following items are specified:

- (i) circulating water flow, L_1 .
- (ii) temperature of water entering the condenser, T_c ; and
- (iii) type of heat rejection device (i.e., mechanized or natural draft tower, or cooling pond).

In the discussion below we indicate how, once the values of L and T_c and the heat rejector type are specified, it is possible to deduce the remaining design characteristics.

Because designs may be characterized by values of L and T_c and heat rejector type, these items play a central role in the design procedure. That is, the alternative designs are those corresponding to alternative combinations of L and T_c and heat rejector type.

Computational Procedure for Recirculation System Designs

The flow chart for the computational procedure is given as Figure A-2. Below we describe the individual steps in the sequence of computations.

1. Set T_c

The values of T_c selected cannot be outside a range of physically meaningful bounds. The lower bound on T_c , designated as T_{cmin} , is computed using

$$T_{cmin} = T_w + \Delta_{min}$$

where values on the right hand side of the equation are input variables and therefore known.

The variable T_c represents the temperature of the circulating water flow leaving the heat rejection device which, for recirculating systems, is equivalent to the temperature of the water delivered to the condenser. Consequently, the lowest temperature it could have (assuming the heat rejection device is an evaporative tower) is the ambient wet bulb temperature. We recognize that the lower bound for T_c will only be approximate for cooling ponds; but rather than add a special set of computations to deal with ponds, we chose to accept this approximation. The procedure for computing the requisite area for ponds contains a special test to eliminate, as technically infeasible, designs which have T_c less than the "equilibrium temperature." However, in design practice the lowest value attainable is considered to be at least a few degrees (i.e., Δ_{min}) above wet bulb. These considerations form the basis for the equation for T_{cmin} above.

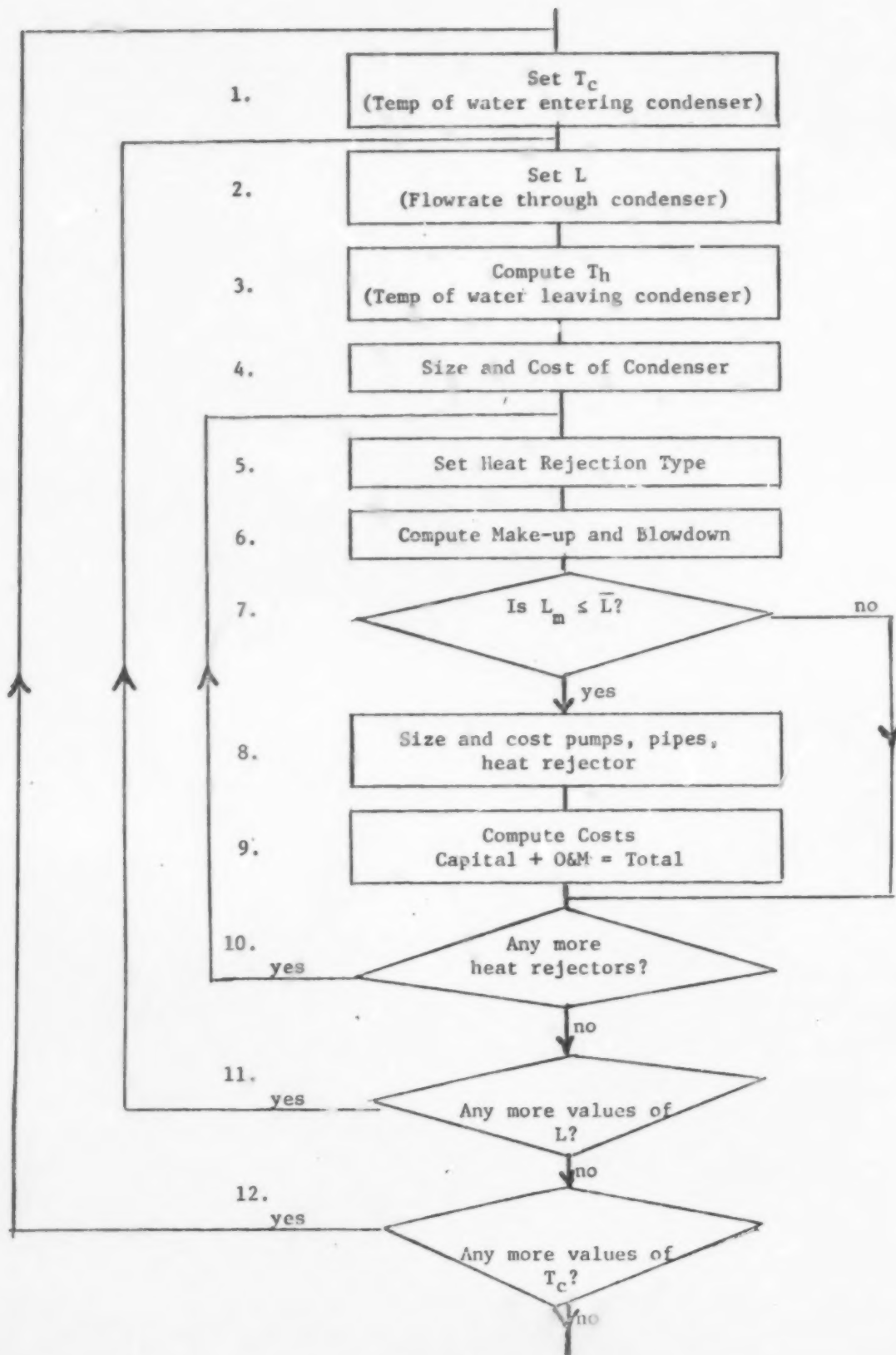


Figure A-2

The upper bound for T_c , designated as $T_{c_{max}}$, is computed using

$$T_{c_{max}} = \text{minimum of } [T_{c_{max1}}, T_{c_{max2}}].$$

The equation for $T_{c_{max1}}$ is

$$T_{c_{max1}} = T_{con} - 5 - R'_{min}.$$

This equation is based on the fact that the hottest possible temperature leaving the condenser is $T_{con} - 5$. (Recall that this is a result of design practice which requires at least a 5°F difference between the temperature of the circulating water leaving the condenser and the temperature of the condensing steam.) Assuming a minimum temperature change of R'_{min} through the heat rejection unit, it follows that the highest possible cold water temperature is given by the equation for $T_{c_{max}}$, above.*

The variable $T_{c_{max2}}$ is not used in the computation of upper bounds on T_c , provided that only a single pressure condenser is employed; i.e., it is introduced only when dual pressure condensers are examined. The computation of $T_{c_{max2}}$ is based on the same considerations employed in deducing $T_{c_{max1}}$, namely, the requirement of a 5°F difference between the temperatures of condensing steam and water leaving the chambers of the condenser.

For dual pressure condensers, let T_{cond} and T_{con} represent the steam condensing temperatures in the first and second chambers, respectively. Observe that the minimum temperature rise across the first chamber is given by $R'_{min}/2$. Therefore, using reasoning parallel to that used in deducing $T_{c_{max1}}$, the value of $T_{c_{max2}}$ is given by

$$T_{c_{max2}} = T_{cond} - 5 - \frac{R'_{min}}{2}$$

The values of $T_{c_{max}}$ and $T_{c_{min}}$ are computed as a preliminary step. Once they are determined, the values of T_c are selected recursively in a manner similar to Step 1 of the procedure for once-through design, previously described.

2. Set L

Before establishing a value for L, the flowrate of the recirculating

*The minimum temperature decrease across the heat rejection unit (R'_{min}) is, in the case of recirculating systems, equal to the minimum temperature increase across the condenser. Both R'_{min} and A_{min} are input parameters determined on the basis of judgment.

water, it is necessary to establish its upper and lower bounds. The upper bound, $L_{\max}$, is given by

$$L_{\max} = \frac{H}{500 R'_{\min}}$$

where $L_{\max}$ is in gal/min, H is in Btu/hour, and $R'_{\min}$ is in $^{\circ}\text{F}$. In other words, the maximum flowrate is determined by the minimum temperature rise through the condenser ($R'_{\min}$).

The lower limit, $L_{\min}$, is given by

$$L_{\min} = \frac{H}{500 (T_{\text{con}} - 5 - T_c)}$$

where the value of T_c corresponds to that set in Step 1. The term $T_{\text{con}} - 5$ represents the maximum possible temperature of cooling water leaving the condenser, and consequently $(T_{\text{con}} - 5 - T_c)$ is the highest possible temperature rise associated with the given value of T_c . The equation gives the minimum flowrate which is associated with this maximum temperature rise.

Having established bounds for the variable L , it is set in a recursive fashion using

$$L = L_{\max} - (j-1) \Delta L_1,$$

where ΔL_1 is an input parameter, j is an index variable on the value of L examined ($j=1, \dots, n_3+1$), and n_3 , the total number of recirculating water flowrates to be considered minus one, is given by the greatest integer in

$$\frac{L_{\max} - L_{\min}}{\Delta L}$$

Using this approach the first value of L considered is $L_{\max}$, the second is $L_{\max} - \Delta L_1$, the third is $L_{\max} - 2\Delta L_1$, etc.

3. Compute T_h

The value of T_h , the temperature of recirculating water leaving the condenser, is found using

$$T_h = T_c + \frac{H}{500L}$$

where T_c is the value set in Step 1, and L is the value set in Step 2. This equation is a result of rearranging

$$L = \frac{H}{500(T_h - T_c)}$$

the fundamental equation which has been used in a number of connections above.

Once the values of T_h and T_c are known, it is a trivial step to compute the "range" as $T_h - T_c$, and the "approach" as $T_c - T_w$; these terms are widely used in the cooling tower industry.

4. Size and cost condenser

The formulas and assumptions used in obtaining estimates of size and capital cost for the condenser are given in subsection A.7.1 below.

5. Set heat rejection type

Our computer program considers three different types of heat rejection devices, in turn, and the index variable i is used to distinguish them. Thus, we use $i = 1$ for mechanical draft cooling tower, $i = 2$ for hyperbolic natural draft cooling tower, and $i = 3$ for cooling pond. This step consists simply of establishing a value for i ($i=1,2$, or 3) or, equivalently, selecting a type of heat rejection unit.

6. Compute make-up and blowdown

Make-up water is the addition to the recirculating water flow required to compensate for water lost by evaporation, windage (or drift), and water released intentionally for purposes of controlling the quality of the recirculating water, commonly referred to as "blowdown" or "purge."

The make-up water requirement was computed from an equation based on the following widely used assumptions [see, for example, reference 4]:

- The evaporative water loss corresponds to one percent of the circulating water flowrate for each 10°F of temperature drop across the heat rejection device.
- A steady state water quality is established whereby the mass of dissolved solids entering the system as make-up will just balance the solids leaving the system as blowdown and wind loss.*

The formula for make-up flow based on these assumptions is [4]:

$$L_m = 0.001 (T_h - T_c) \left(\frac{N}{N-1} \right) L$$

where L_m is the make-up flow, and N the number of "cycles of concentration" is defined as the ratio of dissolved solids in the blowdown to dissolved solids in

*Further details, including the derivation of the equation for computing make-up flow, are given in the Betz Handbook of Industrial Water Conditioning [4].

the make-up; the computer program takes N as an input parameter. The value of N is generally selected on the basis of the concentration of dissolved solids in the make-up water and a judgment regarding the maximum desirable level of solids in the recirculating water.

As an aside, observe that the blowdown flowrate, L_b , which is equivalent to the effluent flowrate, L_e , is computed using [4]

$$L_b = \left[\frac{0.001(T_h - T_c)}{N - 1} \right] L - w_i L$$

where w_i is the windage loss as a fraction of the circulating water flow; this input parameter depends on the heat rejection unit involved and consequently is subscripted with the index, i .

The formulas for make-up water and blowdown implicitly assume that the blowdown flowrate will not be negative (a physically meaningless circumstance). However, since w_i is an input parameter, there is no guarantee that the windage loss will not be so high as to completely eliminate the need for blowdown. To cope with this possibility, we introduce the following test sequence:

If $L_b < 0$, then set $L_b = 0$, and adjust N downward accordingly;
i.e., use

$$N = 1 + \frac{0.001(T_h - T_c)}{w_i}$$

Recompute L_m using this adjusted value of N .

7. Is $L_m \leq \bar{L}$?

Test to see if the make-up flow required is no greater than the maximum allowable diversion, $\bar{L}$. If it is not, the design being considered is not feasible; therefore, go to Step 10. If it is, then establish the identification number associated with this design, note that its effluent characteristics are $L_e = L_b$ and $T_e = T_c$, and continue to Step 8, below.

8. Size and cost pumps, pipes and heat rejection device

The relevant formulas and assumptions are discussed in subsections A.7.2, A.7.3, and A.7.4 below.

9. Compute costs

Capital costs include those for pumps, pipes, heat rejection unit, and condenser. Annual O&M costs are for water supply, if purchased, and treatment, auxiliary power, and maintenance (see subsection A.7.5, below). The total annual cost is the O&M cost plus the fixed charge rate times the capital cost.

At this point, the computations have been completed for the recirculation system design with the following characteristics:

1. T_c , the temperature of water entering the condenser, as set in Step 1;
2. L , the circulating water flowrate, as set in Step 2; and
3. i , the type of heat rejection device, as set in Step 5.

In the remaining steps we change one or more of these characteristics and thereby define a new recirculation system design. The computations above are repeated for this new design in order to obtain details needed to describe the design more completely.

10. Any more heat rejectors?

Test to see if all heat rejector types (i.e., mechanical draft tower, natural draft tower, and cooling pond) have been considered. If they have not, return to Step 5 and proceed with a different type by selecting a different value of i ; otherwise continue.

11. Any more values of L ?

Test to see if the current value of L is the lowest of those to be considered (i.e., have $n_3 + 1$ values of L been examined?). If not, we return to Step 2, get a smaller value of L , and continue with Steps 3, 4, etc.; otherwise we continue with Step 12 below.

12. Any more values of T_c ?

Test to see if the current value of T_c is the lowest to be considered. If not, we return to Step 1, select a smaller value of T_c , and proceed with Steps 2, 3, etc.; otherwise, the design procedure for recirculating system is complete.

The number of designs generated by the step-by-step procedure above is equal to the number of different combinations of T_c , L and i considered. We introduce the following example to further illustrate the manner in which alternative designs are generated.

Consider a case where a single pressure condenser is employed and the value of T_{con} is set at 108°F ; suppose, further, that $T_w = 78^\circ\text{F}$, $\Delta T_2 = 3^\circ\text{F}$, $R'_{min} = 10^\circ\text{F}$ and $A_{min} = 8^\circ\text{F}$. The number of different values of T_c examined is equal to $(n_2 + 1)$ where

$$n_2 = \left[\frac{T_{cmax} - T_{cmin}}{T_2} \right]$$

As indicated in the discussion of Step 1 above, the values of $T_{c\max}$ and $T_{c\min}$ are determined using

$$T_{c\max} = T_{c\max 1} = T_{con} - 5 - R'_{min} = 108 - 5 - 10 = 93, \text{ and}$$

$$T_{c\min} = T_w + A_{min} = 86.$$

Therefore,

$$\eta_2 = \left[\frac{93 - 86}{3} \right] = 2$$

Thus, there will be three values of T_c considered, namely, 93, 90 and 87°F. The number of different values of L examined depends on the value of T_c under consideration. Recall from Step 2 above that for a given value of T_c , there will be $(\eta_3 + 1)$ alternative flowrates examined, where

$$\eta_3 = \left[\frac{L_{\max} - L_{\min}}{\Delta L} \right],$$

$$L_{\max} = \frac{H}{500 R'_{min}}, \text{ and}$$

$$L_{\min} = \frac{H}{500 (T_{con} - 5 - T_c)}.$$

To continue the example, suppose $H = 2 \times 10^9$ Btu/hr and $\Delta L = 50,000$ gpm. The maximum feasible value of L , which is independent of T_c , is

$$L_{\max} = \frac{2 \times 10^9}{(500)(10)} = 400,000 \text{ gpm.}$$

The minimum feasible value of L depends on T_c . For the case where $T_c = 93$, the minimum is

$$L_{\min} = \frac{2 \times 10^9}{(500)(108 - 5 - 93)} = 400,000 \text{ gpm.}$$

The value of η_3 is obviously zero, and the number of flowrates (i.e., $\eta_3 + 1$) is one; this flowrate corresponds to the only feasible value, namely, 400,000 gpm.

For the case where $T_c = 90^\circ\text{F}$,

$$L_{\min} = \frac{2 \times 10^9}{(500)(108 - 5 - 90)} = 308,000 \text{ gpm.}$$

The corresponding value of η_3 is

$$\eta_3 = \left[\frac{400,000 - 308,000}{50,000} \right] = 1.$$

In this case there would be two flowrates, namely, $L = 400,000$ gpm and $L = 350,000$ gpm. Repeating the computation for $T_c = 87^\circ\text{F}$, it is easy to show that there would be four different flowrates, beginning with 400,000 gpm, and decreasing in increments of 50,000, down to 250,000 gpm.

Assuming that each of three types of heat rejection unit would be considered for the alternative combination of L and T_c , there would be a total of 21 recirculation system designs examined. The alternative combinations of T_c , L and i are summarized in the table below.

Values of T_c ($^\circ\text{F}$)	Values of L (10^3 gpm)	Values of i (Index of Heat Rejector Type)	Number of Alternative Combinations
93	400	1,2,3	3
90	400,350	1,2,3	6
87	400,350,300,250	1,2,3	12
			21 Total

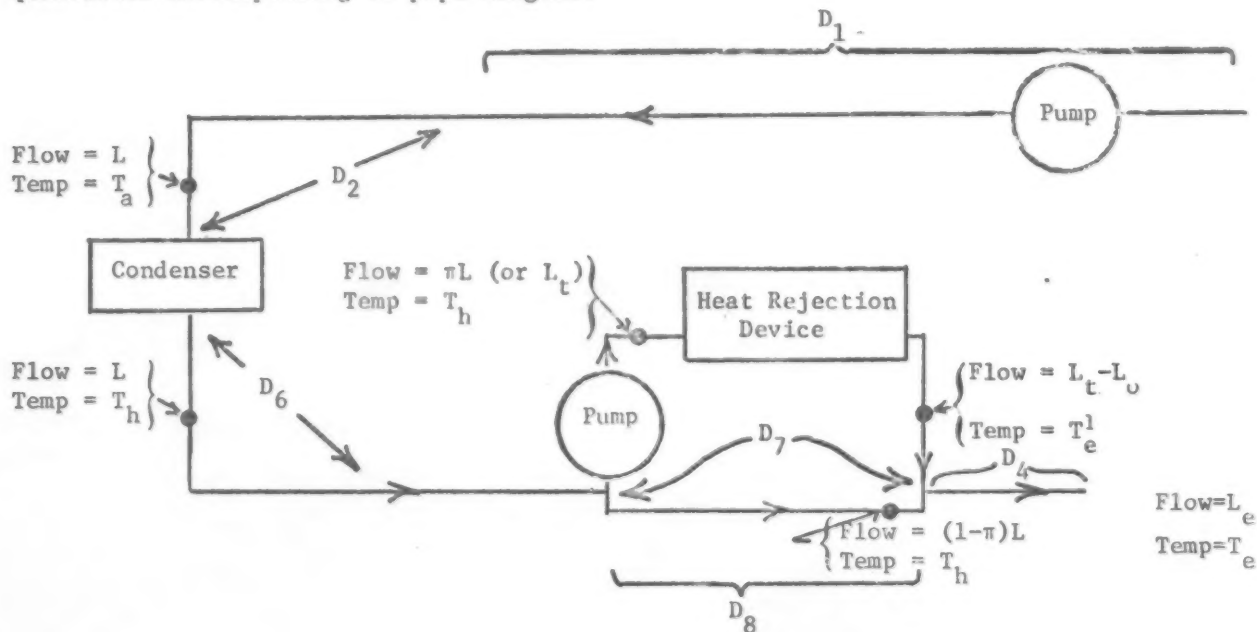
A.5 Helper System

(i) Pipe Layout and Other System Components

A helper system, as the term is used herein, is essentially a once-through system modified to permit the "treatment" (i.e., cooling) or part or all of the outflow from the condenser prior to its discharge to a receiving water. A helper system consists of the following elements:

- Intake and discharge pipes,
- A condenser,
- A heat rejection device,
- A pump capable of meeting all requirements for pumping except those directly associated with the heat rejection unit,
- A pump to achieve the head requirements associated with the particular heat rejection system.

The diagram below shows the pipe layout; as before, the variables D_i are input quantities corresponding to pipe lengths.



Of the total flow of L gal/min leaving the condenser, a quantity of πL is cooled from T_h down to T_c^1 . This cooled water is then mixed with $(1-\pi)L$ at T_h to yield an effluent with a temperature T_e , somewhere between T_c and T_h depending on the value of π .

Alternative helper designs include different combinations of the following:

- (i) L , Intake flowrate;
- (ii) π , Fraction of intake flowrate diverted to heat rejection unit;
- (iii) Type of heat rejection unit; and
- (iv) T_i , Temperature at discharge from heat rejection unit.

Once these values are specified, it is possible, using standard engineering formulas,

to compute other design characteristics, including the effluent temperature and flowrate, condenser area, pipe diameters, etc.

(iii) Computational Procedure for Helper System Designs

1. Set L , the flowrate through the condenser.*

The maximum possible value of the intake flowrate, L , is given by

$$L_{\max} = \text{minimum of } \left[\bar{L}, \frac{H}{500 R_{\min}} \right].$$

The interpretation of the first bracketed term is clear, since $L_{\max}$ cannot possibly be greater than the maximum allowable diversion $\bar{L}$. The second bracketed term, $H/500 R_{\min}$, represents the flowrate associated with the smallest possible temperature rise for water flowing through the condenser. If this term is smaller than $\bar{L}$, then it is set equal to $L_{\max}$; otherwise $\bar{L}$ is set equal to $L_{\max}$.

The minimum value of the intake flowrate is given by

$$L_{\min} = \frac{H}{500 (T_{\text{con}} - 5 - T_a)}.$$

The explanation of this equation has been given in the discussion of Step 2 of the procedure for developing alternative recirculation system designs. Once the upper and lower bounds on L are computed, we perform a simple test to be certain that $L_{\max} \geq L_{\min}$. If it is not, then there are no technically feasible helper designs consistent with the postulated conditions, and the step-by-step procedure is terminated. If $L_{\max} \geq L_{\min}$, then we proceed to establish alternative values of L in much the same manner as described in Step 2 of the recirculating system design procedure.

2. Compute T_h , the temperature of water leaving the condenser.

Using L set in Step 1, and T_a fixed by ambient conditions, it is straightforward to compute T_h using

$$T_h = T_a + \frac{H}{500 L}.$$

3. Set T_c^1 , the temperature of water leaving the heat rejection unit.

The highest possible value for T_c^1 is given by

$$T_{c\max}^1 = T_h - R_{\min}^1.$$

The right-hand side is simply the temperature of the entering water (as computed in Step 2) minus the minimum rise, $R_{\min}^1$.

* Since L and T_h are related, an alternative way to begin the procedure involves setting T_h , and then computing L .

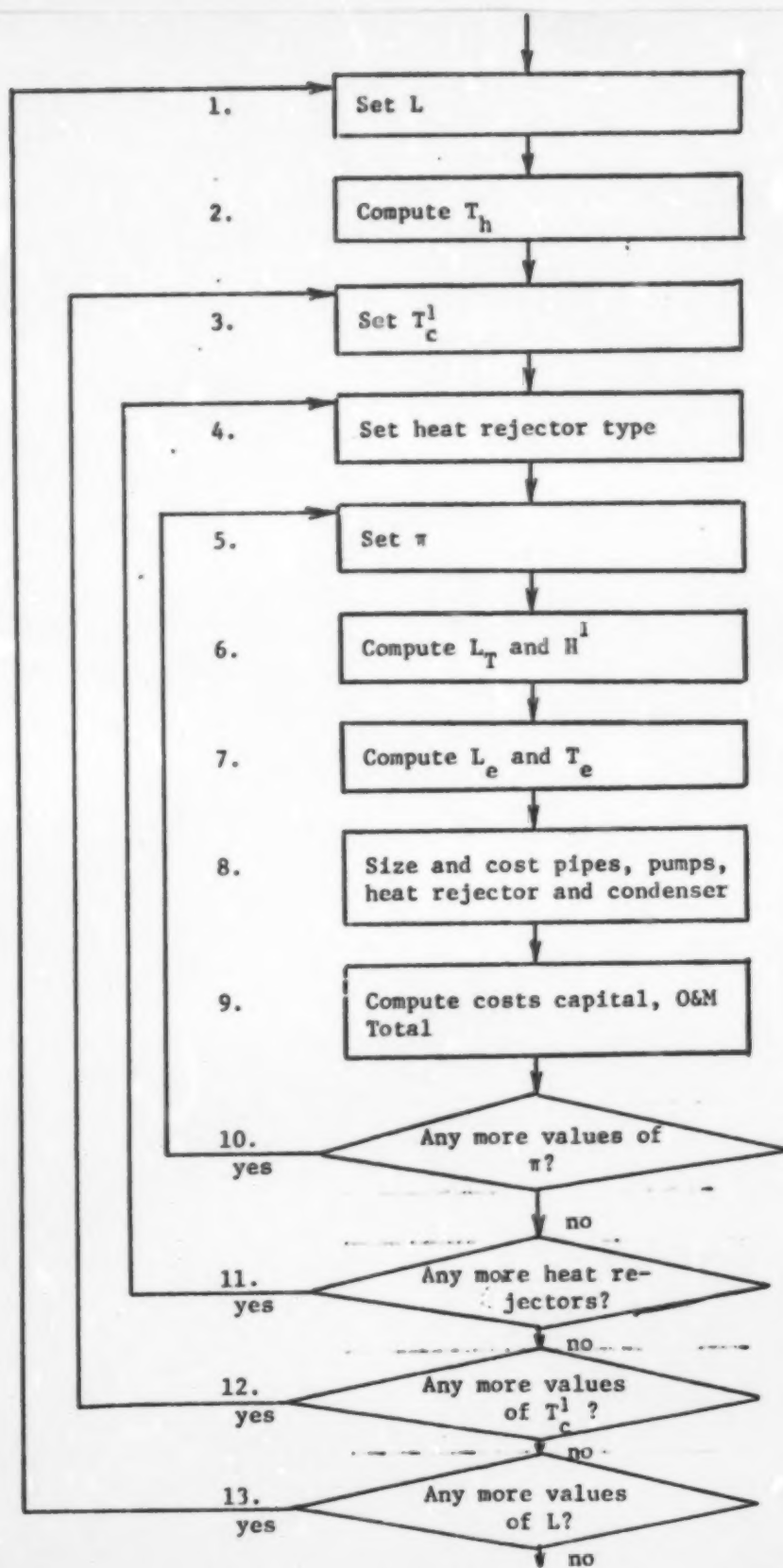


Figure A-3

Flow Chart for Helper System Design Computations

The lowest possible value for T_c is

$$T_{c \min}^1 = T_w + A_{\min},$$

where $A_{\min}$, an input parameter, is the minimum approach to be considered.

Observe that, as in the case of flowrates examined in Step 1, we perform a simple test before proceeding, to be certain that $T_{c \max}^1 \geq T_{c \min}^1$. If it is not, then there are no technically feasible designs consistent with the value of L set in Step 1. We proceed by returning to Step 1, selecting a lower value of L , and continuing to Step 2, etc.

If $T_{c \max}^1 \geq T_{c \min}^1$, then the values of T_c^1 are selected recursively in a manner similar to Step 1 of the procedure for once-through system designs previously described.

4. Set heat rejector type

This step consists of selecting the type of heat rejector to be employed.*

5. Set Π

We establish a host of values of Π using

$$\Pi = \frac{k}{p}, \text{ where } k = 1, 2, \dots, p,$$

and p is an input parameter that represents one minus the number of equally spaced values of Π to be considered. From the way in which Π is defined it clearly must be such that $0 < \Pi \leq 1$. The equation above has $\frac{1}{p}$ as the first value of Π , $\frac{2}{p}$ as the second, all the way to one as the last value of Π .

Once a value of Π is selected, it is necessary to establish an identification number for the design under consideration. This simply serves to keep track of the characteristics of a given design.

6. Compute L_T and H'

The flow to the heat rejection unit is, by definition,

$$L_T = \Pi L.$$

The associated heat rejected, given a flowrate of L_T , an inflow temperature of T_h (from Step 2), and an outflow temperature of T_c^1 (from Step 3) is

$$H' = 500 L_T (T_h - T_c^1)$$

Observe also that knowing T_h and T_c^1 it is possible to compute the "range," $T_h - T_c^1$, and the "approach" $T_c^1 - T_w$, which are both essential to the determination of costs for the heat rejection unit.

*In the computer program, we limited the heat rejector types to mechanical draft towers and cooling ponds. It is unlikely that natural draft towers would be considered for the relatively low heat rejection rates associated with some helper designs.

7. Compute L_e and T_e

Before computing the effluent flowrate it is necessary to account for evaporative and windage losses for flow through the tower. Evaporative losses, assuming a loss of 1 percent of flow per 10°F drop in temperature, are simply

$$0.001 (T_h - T_c^1) L_T$$

The total loss, denoted as L_o , is

$$L_o = 0.001 (T_h - T_c^1) L_T + w_i + L_T$$

where w_i is the wind loss parameter associated with the heat rejection device under consideration. (If it is a mechanical tower, $i = 1$, etc.)

The effluent flowrate is the flow through the heat rejection unit, less the corresponding losses, L_o , plus the flow that did not go through the heat rejection unit; i.e.

$$L_e = L_T - L_o + (L - L_T), \text{ or}$$

$$L_e = L - L_o.$$

The effluent temperature is the flow weighted average for the two streams; i.e.,

$$T_e = \frac{(L_T - L_o) T_c^1 + (L - L_T) T_h}{L_e}$$

8. Size and cost pipes, pumps, heat rejection device, and condenser

The relevant formulas and assumptions are discussed in subsections A.7.1 through A.7.4, below.

9. Compute costs

Capital costs include those for pumps, pipes, heat rejection device and condenser. Annual O & M costs are for water supply, if purchased, and treatment, auxiliary power, and maintenance.* (See subsection A.7.5, below.) The total annual cost is the O & M cost plus the product of the fixed charge rate times the capital cost.

*In computing the O&M cost associated with the tower or pond the number of hours per year of plant operation is multiplied by a known constant between zero and one. This constant reflects the fact that such a heat rejection device might be used for only part of the year.

10. Any more values of Π ?

Test to see if the value of Π is equal to one. If it is, we continue; otherwise, we go to Step 5, obtain a larger value of Π , and proceed through Steps 6, 7, etc.

11. Any more heat rejectors?

Test to see if all heat rejector types have been considered. If they have not, we return to Step 5 and proceed with a different type; otherwise we continue.

12. Any more values of T_c^1 ?

Test to see if the current value of T_c^1 is the lowest of those to be considered. If it is, we continue; otherwise we go to Step 4, get a smaller value of T_c^1 , and proceed to Steps 5, 6, etc.

13. Any more values of L ?

Test to see if the current value of L is the lowest to be considered. If it is not, we go to Step 1, get a smaller value of L , and proceed to Steps 2, 3, etc. If L is the lowest value to be considered, then the design procedure for helper systems is complete.

The alternative helper system designs are essentially defined by four characteristics, namely, L , T_c^1 , Π and the heat rejector type. As in the discussion of recirculation systems, we use a hypothetical example to further illustrate the nature of the alternative design.

Consider a case involving a single pressure condenser with $T_{con} = 108^\circ\text{F}$ and other input parameters specified as follows: $H = 2 \times 10^9$ Btu/hr, $T_a = 77^\circ\text{F}$, $T_w = 78^\circ\text{F}$, $R_{min} = 2^\circ\text{F}$, $R_{min}^1 = 10^\circ\text{F}$, $\Delta_{min} = 8^\circ\text{F}$, $\Delta T = 3^\circ\text{F}$, $\Delta L = 50,000$ gpm and $\bar{L} = 400,000$ gpm. The upper bound on flowrate is

$$L_{max} = \text{minimum of } \left[\bar{L}, \frac{H}{500 R_{min}} \right]$$
$$L_{max} = \text{minimum of } \left[400,000, \frac{2 \times 10^9}{500 (2)} \right] = 400,000 \text{ gpm}$$

The lower bound on L is

$$L_{min} = \frac{H}{500 (T_{con} - 5 - T_a)} = \frac{2 \times 10^9}{500 (108 - 5 - 77)} = 154,000 \text{ gpm}$$

The alternative flowrates to be examined are determined by first computing

$$\left[\frac{L_{max} - L_{min}}{\Delta L} \right] = \left[\frac{400,000 - 154,000}{50,000} \right] = 4.$$

Therefore, five flowrates are to be considered, beginning with 400,000 gpm, and decreasing in increments of 50,000 gpm, down to 200,000 gpm.

In the case where $L = 400,000$, the associated value of T_h is

$$T_h = T_a + \frac{H}{500 L} = 77 + \frac{2 \times 10^9}{500 \times 400,000} = 87.$$

the value of T_c^1 max is

$$T_{c \max}^1 = T_h - R_{\min}^1 = 87 - 10 = 77,$$

and T_c^1 min is

$$T_{c \min}^1 = T_w + \Delta_{\min} = 78 + 8 = 86.$$

Since the value of $T_{c \max}^1 < T_{c \min}^1$, we conclude that there is no design consistent with the postulated conditions. Repeating the above computation shows that this inconsistency holds for all values of L down to $L = 200,000$ gpm. In this latter case, the computed value of $T_h = 97$; also, $T_{c \max}^1 = 87$ and $T_{c \min}^1 = 86$.

To determine the alternative values of T_c^1 to be examined, we first compute

$$\left[\frac{T_{c \max}^1 - T_{c \min}^1}{\Delta T} \right] = \left[\frac{87 - 86}{3} \right] = 0.$$

Therefore, there is but a single temperature to be examined, namely, $T_c^1 = 87$.

Suppose the value of p in input at 3. This implies that the values of Π considered are $1/3$, $2/3$ and 1 . For each of these values of Π we consider two heat rejector types (mechanical draft towers and cooling ponds), yielding < six different designs in total. Observe that the values of $R_{\min}^1$ and $\Delta_{\min}^1$ play an important role in restricting the number of designs to be examined.

A.6 Deducing Minimum Cost Designs

The preceding three sections described the manner in which individual designs of various types were developed. Figure A-4 sketches, in general terms, the flow of computations in the portion of the design procedure described thus far. It serves to put the detailed discussions of the preceding four sections in perspective. The figure also describes the computer program capacity to selectively examine only once-through designs, or once-through and helper but not recirculation designs, etc.; this is accomplished by changing various program instruction variables.

The portion of the design procedure described thus far consists of obtaining all designs that are to be developed. For each specific design, all major design details are stored as are the capital and O & M costs for various items; for purposes of deducing the minimum cost cooling system design, the essential stored information consists of a design identification number, the associated costs, and the effluent characteristics. More precisely, consider that the following information is stored for each design:

k = design identification index;

C_k = total annual cost for design k ;

L_{ek} = effluent flowrate for design k ; and

T_{ek} = effluent temperature for design k .

With this information available it is not at all difficult to isolate the set of all designs capable of meeting a specified effluent standard, given in terms of L^* and T^* , where

L^* = maximum allowable effluent flowrate, and

T^* = maximum allowable effluent temperature.

Such a set of designs is defined by all values of k that satisfy the following constraints:

$$L_{ek} \leq L^*, \text{ and}$$

$$T_{ek} \leq T^*.$$

Let the group of values of k meeting these constants be collectively referred to as the set K . The cooling system design meeting the standard given by L^* and T^* , is defined by the value of k from the set K having the lowest cost. That is, if the index associated with the minimum cost design is denoted k^0 , then k^0 is such that

$$C_{k^0} \leq C_k$$

for all $k \neq k^0$ in the set K .

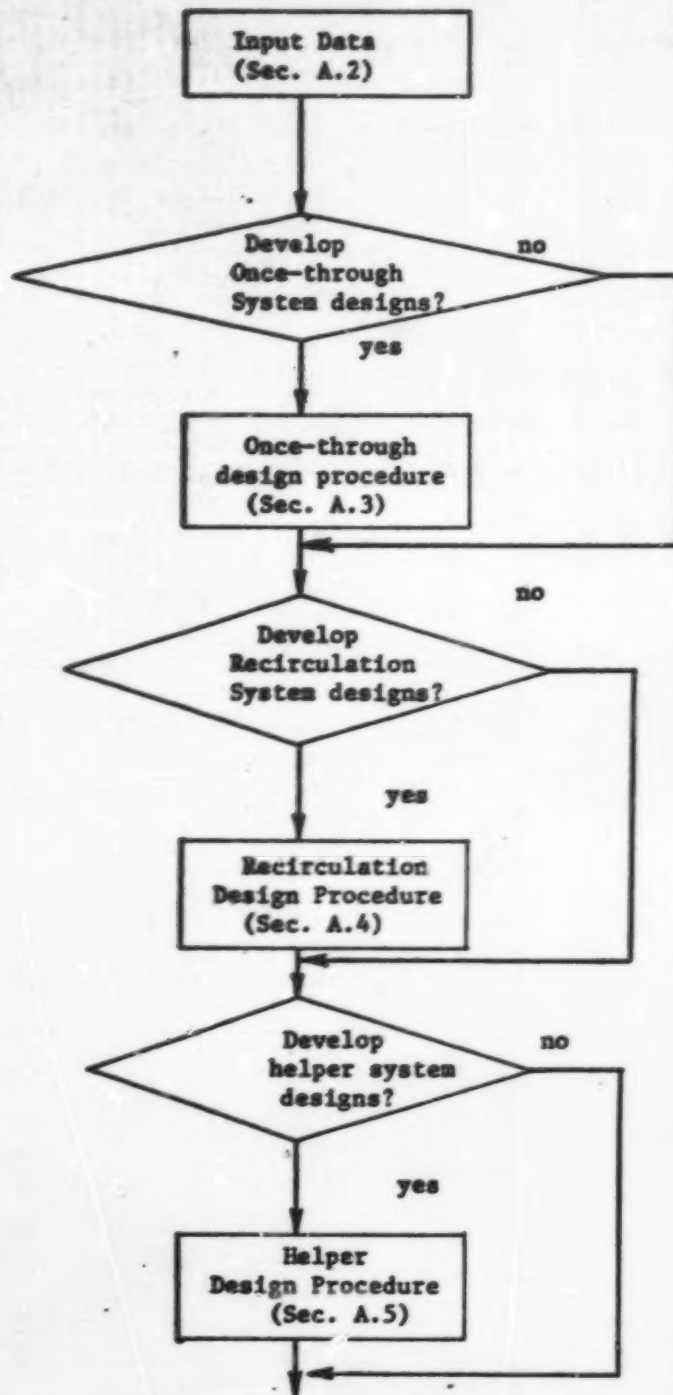


Figure A-4

Flow Chart for Generation of Alternative Designs

The process of isolating designs meeting a given standard, and selecting the minimum cost design from among those isolated, is a routine operation using a digital computer. In fact, the computer program is arranged such that a wide array of combinations of L^* and T^* are considered in sequence. This leads to the generation of relationships which show the costs of the least costly cooling system to meet a given standard as a function of the standard (i.e., L^* and T^*). This procedure was followed in developing the results presented in the main body of this report.

Recall from Section A.1 that the procedure for generating the least costly cooling system design to meet a given thermal quality standard is not an end in itself. The entire cooling system design procedure is repeated a number of times, once for each alternative combination of plant design capacity, steam cycle design, and plant utilization schedule. In other words, the procedure described above must be viewed as a single component of the broader analysis presented in Section I of the main report.

The remainder of this Appendix is devoted to a presentation of the assumptions and functional relationships used in connection with the following aspects of our computer program design: cost and sizing of condenser, heat rejection units, pipes, and pumps; and annual operation and maintenance costs.

A.7 Assumptions and Formulas Used in Design of Individual Cooling System Components

A.7.1 Condenser

In a steam electric power plant the condenser serves to transfer heat from the turbine exhaust steam to the cooling water. For plants in the United States "shell and tube" condensers are used, and the condensing process is often carried out with the steam held at a constant pressure. We discuss the formulas and assumptions used for this single pressure condenser case first. Following this, we discuss the way these formulas can be modified to deal with dual pressure condensers.

In the context of the design procedure described in Section A.2-A.5 above, there are two groups of quantities taken as given for purposes of condenser design. Quantities set in an iterative fashion within the individual design portion of the program include:

L = cooling water flowrate (gal/min),

T_c = inlet (i.e., cold) water temperature ($^{\circ}\text{F}$), and

T_h = outlet (i.e., hot) water temperature ($^{\circ}\text{F}$).

Quantities taking the form of parameters "input" to the computer include:

T_{con} = Temperature of condensing steam ($^{\circ}\text{F}$)

H = condenser duty (Btu/hr),

V = cooling water velocity (ft/sec),

d = diameter of condenser tubes (inches),

a_c, b_c, γ_c = constants in capital cost function, and

K_1 = an integer representing the number of passes through the condenser.

The condenser design procedure involves the determination of (i) the required surface area, (ii) the associated tube length, (iii) the capital cost, and (iv) the head loss through the condenser. We discuss each of these computations below.

(i) Surface Area

The condenser area computation is performed using the following standard formula in engineering design practice [Ref. 16, p. 349]:

$$A_s = \frac{H}{u \Delta T_m}$$

where

A_s = surface area (ft^2),

u = heat transfer coefficient (Btu/hr/sq.ft./ $^{\circ}\text{F}$), and

ΔT_m = log mean temperature difference.

The value of u , the heat transfer coefficient, is computed as

$$u = C_1 [1 + 0.008(T_c - 70)]$$

where C_1 (explained below) is an input parameter, and the bracketed term represents a correction factor to account for the influence of variations in the temperature of circulating water entering the condenser (T_g). The bracketed term is a linear approximation to a nonlinear correction curve presented in the Standards for Steam Surface Condensers [Ref. 13, p. 5].

The input parameter C_1 accounts for the influence of several factors which are considered as constants in the context of the design procedure: i.e.,

$$C_1 = s_1 s_2 s_3 \sqrt{V},$$

where

s_1 = correction factor for tube material and gauge;
typically $0.5 \leq s_1 \leq 1.0$, [see Ref. 1, p. 19].

s_2 = constant depending on tube diameter; typically
 $263 \leq s_2 \leq 270$, [Ref. 1, p. 7].

s_3 = "cleanliness factor"; often assumed to be 0.85, [Ref. 1, p. 6].

V = velocity (ft/sec); typically $5 \leq V \leq 8$.

The formulas above for u and C_1 are standard for the design of condensers used in steam power plants.

The value of ΔT_m in the condenser area formula above is the "log-mean-temperature-difference" defined as

$$\Delta T_m = \frac{T_h - T_c}{\ln \left[\frac{T_{con} - T_c}{T_{con} - T_h} \right]}.$$

(11) Tube Length

The equation for computing tube length is easily deduced by first writing expressions for A_x , the cross sectional area of all tubes, and n_c , the total number of tubes. The cross sectional area is simply

$$A_x = \frac{\left(\frac{L}{449} \right)}{V},$$

where the constant 449 is the number of gallons per minute in one cubic foot per second and A_x is in square feet. This equation is a form of the law of conservation of mass commonly used in elementary fluid mechanics.

The total number of tubes, n_c , is found by dividing A_x by the cross sectional area of a single tube; i.e.,

$$n_c = \frac{A_x}{\pi \left(\frac{d_c^2}{24} \right)} = \frac{576 A_x}{\pi d_c^2}$$

where d_c (the inside diameter of condenser tubes) is divided by 2 to convert to radius, and 12 to convert inches to feet.

Given the total number of tubes, we can determine the length of each tube needed to provide A_s square feet of surface area. The requisite tube length, D_c , is found by dividing A_s by the total outside surface area per foot of tube; i.e.,

$$D_c = \frac{A_s}{2\pi \left(\frac{d_c^1}{24} \right) n_c} = \frac{3.82 A_s}{d_c^1 n_c}$$

where D_c is in units of feet and d_c^1 is the outside diameter of condenser tubes in inches.

After substituting the formulas above for A_x and n_c into the equation for D_c , the latter simplifies to

$$D_c = 9.35 \frac{V d_c^2 A_s}{d_c^1 L}$$

(iii) Capital Cost

In developing a cost estimating equation for condensers, we relied on price lists supplied by a major condenser manufacturer. Condenser costs were estimated by deducing a base cost for the shell and standard auxiliaries (assuming a tube length of 30 feet). Using the manufacturers price list data, we estimated this as a linear function of A_s ; i.e.,

$$\alpha_c A_s + \beta_c,$$

where $\alpha_c = 1.2$ and $\beta_c = 75,000$ are the values of the constants obtained in fitting the price list data.

Account is taken of the costs associated with tube lengths other than 30 feet by multiplying the base cost by the linear correction factor

$$0.015 D_c + 0.55.$$

To obtain a final estimate it is necessary to add tube costs to the shell costs. We estimated the cost for tubes using

$$\gamma_c n_c D_c,$$

where γ , the cost per foot of tube, is an input parameter to the computer program. This linear form was deduced on the basis of price lists supplied by a condenser tube manufacturer.

Bringing these various terms together, the total uninstalled condenser cost is

$$[0.015 D_c + 0.55] (\alpha_c A_s + \beta_c) + \gamma_c n_c D_c.$$

We estimated the total cost using this expression plus a fraction of it to account for the cost of installation.

(iv) Head Loss

The loss of head for the circulating water passing through the condenser is computed in two parts. The first part, the pressure loss in water boxes, is a nonlinear function of velocity, V as presented by the Baldwin-Lima-Hamilton Corporation [Ref. 1], and the Heat Exchange Institute [Ref. 13]. We used a piece-wise linear approximation to the standard pressure loss function [Ref. 1, p. 21]. Thus, for $V \leq 4.5$, the pressure loss is

$$0.20 V - 0.18,$$

and for $V > 4.5$, the pressure loss is

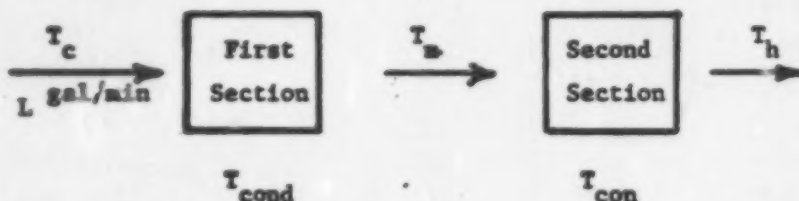
$$0.29 V - 0.578.$$

If K_1 , the number of passes, is not equal to one, then the pressure loss is taken as K_1 times the appropriate expression above.

The second part of the head loss is that attributable to friction losses associated with flow through the tubes. These losses are computed using a variation of the Hazen-Williams equation and are represented by [Ref. 1, p. 23].

$$0.0067 \frac{V^{1.84} D_c}{d_c^{1.16}}.$$

The formulas above summarize the procedure employed in the case of single pressure condensers. We now discuss the modifications required to deal with a dual pressure condenser. We assume that the condenser duty will be split equally between the two sections, and we let T_{cond} and T_{con} (where $T_{\text{cond}} < T_{\text{con}}$) represent the given steam condensing temperatures associated with each section. The sketch below summarizes the terminology employed.



It follows from the assumption concerning the split in condenser duty that the temperature between sections, T_m , is

$$T_m = T_c + 1/2 (T_h - T_c) = \frac{T_c + T_h}{2}.$$

The modifications to the single pressure condenser design procedure relate to the computation of A_s . For the dual pressure case A_s is computed as the sum of A'_s and A''_s , where

$$A'_s = \frac{H/2}{u \Delta T'_m}, \text{ and}$$

$$A''_s = \frac{H/2}{u \Delta T''_m}.$$

The requisite values of log-mean-temperature-difference are given by

$$\Delta T'_m = \frac{T_m - T_c}{\ln \frac{T_{\text{cond}} - T_c}{T_{\text{cond}} - T_m}}, \text{ and}$$

$$\Delta T''_m = \frac{T_h - T_m}{\ln \frac{T_{\text{con}} - T_m}{T_{\text{con}} - T_h}}.$$

A.7.2 Heat Rejection Units

Three different types of heat rejection units have been included in our computer program: mechanical draft cooling towers, natural draft cooling towers, and cooling ponds. Below we outline the assumptions and formulas used in estimating design parameters and associated installed costs.

Mechanical Draft Towers

There are a variety of classifications for mechanical draft cooling towers depending, for example, on the location of the fans (forced vs induced draft) and the relative directions of air and water flow (counterflow vs crossflow). For power plants in the United States the induced draft crossflow type tower has been used widely and consequently it is used herein as a representative mechanical tower [Ref. 9].

The procedures used to obtain estimates of tower size and cost are developed, in part, from a presentation by Dickey and Cates [Ref. 9]. They define the parameter "required tower units," TU, as

$$TU = (RF) (L)$$

where L is the flowrate of water through the tower and RF is a "rating factor" which is meant to indicate the extent to which it is more or less difficult to accomplish cooling under given design conditions as compared to a selected base condition. These conditions essentially involve the following parameters:

T_w = wet bulb temperature ($^{\circ}F$)

R = range (i.e., the difference between the temperature of water entering and leaving the tower)

A = approach (i.e., the difference between the temperature of the water leaving the tower and T_w).

Dickey and Cates present a series of curves which permit the derivation of the rating factor for a selected induced draft crossflow tower, given the design values of range, approach and wet-bulb temperature. As discussed in Appendix B, below, we employed a statistical procedure to reduce the graphical approach to one involving an equation. A typical form for the equations developed is

$$RF = c_m R^{r_m} A^{a_m} T_w^{w_m}$$

where c_m , r_m , a_m , and w_m are constants.

Dickey and Cates also presented data on the actual budgeted costs for eight different towers (with flowrates greater than 100,000 gpm) that have already been designed and are scheduled for construction by 1972. These data indicated that the unit costs expressed as dollars per tower unit (α_m) were not widely scattered; they had a mean value of 4.91 and a standard deviation of 0.57. Consequently, in estimating costs we simply multiplied the unit cost by the value of TU.

The costs computed using the data presented by Dickey and Cates include the purchase and installation of the complete cooling tower structure, filling, fans, and motor.^{*} However, the cost of the cold water basins and fan motor hookups are omitted. Power plant designers have estimated the cost of the cold water basins over a rather broad range from a low of about \$3 per square foot of basin to a high in excess of \$8 per square foot; we used an intermediate value of \$6 per square foot. The requisite cooling basin area was estimated using an admittedly approximate procedure suggested by McKelvey and Brooks [Ref. 22]. This involved dividing the flowrate through the tower by the water packing load which we took as 5 gpm/square foot. Fan motor hookup charges were estimated by multiplying the fan horsepower by a unit cost of \$25 per horsepower.^{**} Following Cootner and Lof we assumed the fan horsepower to be equal to one percent of the product of the rating factor and flowrate (in gpm) [Ref. 7, p. 69].

The design procedure for mechanical towers may be summarized as follows. Given the design range, approach and wet bulb temperature, compute the rating factor using equations derived in Appendix B; and evaluate the number of required tower units, and the costs. This is acknowledged to be a drastic oversimplification of the rather detailed design and cost analyses that would be performed by cooling tower manufacturers; nevertheless, it should provide a useful cost approximation consistent with the level of detail employed in other portions of this study.

Natural Draft Towers

Natural draft towers of the hyperbolic type are playing an increasingly important role in the electric utility industry. Although both crossflow and counterflow towers are used in the United States, the computations in this study are for the crossflow type; this choice was made solely on the basis of the availability of cost information.

^{*}Cates, R.E., The Marley Company, Kansas City, Mo., personal communication, 4/7/71

^{**}Tiechnor, B., Environmental Protection Agency, Corvallis Oregon, personal communication, March 5, 1971.

Dickey and Cates have presented "typical cost-performance curves" for hyperbolic natural draft crossflow towers [Ref. 9]. These curves allow the computation of the unit cost of the tower (\$/10³Btu/hr) given the design range, approach wet bulb temperature and relative humidity. These costs include the cold water basin.* As discussed in Appendix B, below, we employed a statistical procedure to reduce the curves to an equation.

For natural draft towers we employed such equations to compute \$/10³Btu/hr given the values of range, approach, wet bulb temperature and relative humidity. The total installed cost of the tower is obtained by multiplying this unit by the condenser duty in units of 10³ Btu/hr.

Cooling Ponds

The area for cooling ponds is often determined by established rules of thumb, as for example, the often quoted one acre per megawatt figure for fossil fueled plants. However, research efforts on heat transfer from ponds should lead to more sophisticated design procedures which permit accounting for local climatologic features (e.g., wind speed, solar radiation, etc.). While there is certainly no single heat transfer model currently regarded as appropriate under all circumstances, we felt it useful to use a single model as a matter of computational convenience, and for ease in interpreting results.

The computations for cooling pond area are based on the heat transfer model described by Edinger and Geyer for "flow through" (or, in the language of chemical engineers, "plug flow") pattern of fluid motion. Edinger and Geyer demonstrated that under postulated plug flow conditions it is possible to relate the pond inflow and outflow temperatures to pond surface area using the following equation [Ref. 10, pp. 106-122]:

$$\frac{T_c - E}{T_h - E} = \exp \left[- \frac{K A'}{62.4 L'} \right]$$

where:

- T_c = outflow (or cold water) temperature (°F);
- T_h = inflow (or hot water) temperature (°F);
- E = a parameter representing "equilibrium" temperature (°F);
- K = heat transfer coefficient (Btu/ft²-day-°F);
- A' = pond surface area (sq. ft); and

* R.E. Cates, The Marley Company, Kansas City, Mo., personal communication, April 7, 1971.

L' = circulating water flowrate (cu.ft/day).

This equation may be rearranged to give pond surface area in terms of all other parameters, which in the procedure followed herein, are known values. The result of this algebraic manipulation is

$$A' = - \frac{62.4 L'}{K} \left[\ln \frac{T_c - E}{T_h - E} \right]$$

This equation, when modified to contain units consistent with those employed above is

$$A = - \frac{(62.4)(193)}{43,560} \left(\frac{L}{K} \right) \left[\ln \left(\frac{T_c - E}{T_h - E} \right) \right]$$

where A is pond surface area in acres, L is flowrate through the pond (i.e., the circulating water flowrate) in gallons per minute, 193 is the number of cubic feet of water per day in one gallon per minute, and 43,560 is the number of sq. ft per acre.

The values of T_c , T_h , and L are fixed by the nature of the cooling system design being examined. The values of K and E are input parameters that can be determined using climatological data available in standard references [Ref. 20], and a computational procedure due to Brady *et al.* [Ref. 5].

The procedure for computing E and K is an iterative one requiring that the following design parameters be known:

- T_d = dew point temperature ($^{\circ}\text{F}$)
- S = daily solar radiation ($\text{Btu/ft}^2\text{-day}$)
- W = windspeed (miles/hour)*
- $\hat{E}$ = initial estimate of E ($^{\circ}\text{F}$)

The equations to be solved simultaneously for E and K are as follows [Ref. 5]:

$$T = \frac{E + T_d}{2}$$

$$B = 0.255 - 0.0085T + 0.000204 T^2$$

* Values of T_d , S, and W for various parts of the U.S. are summarized in the Climatic Atlas of the United States [Ref. 20].

$$K = 15.7 + (8 + 0.26) (70 + 0.7 W^2)$$

$$E = T_d + \frac{S}{K}$$

Details of the trial and error solution procedure as well as the physical basis for the equations are given in Brady et al. [Ref.5, Chapter 7]. Once the values of E and K are determined, it is possible to compute A, the required pond area, using the formula above. The total cost directly associated with the cooling pond is $a_p A$, where a_p is the unit cost for land and pond construction in \$/acre.

A.7.3 Pipes

In developing the computer program, we assumed that water conveyance within the cooling system is accomplished exclusively by means of pipes. We recognize, however, that there may be circumstances in which local conditions may dictate the use of open channels in a portion of the system (e.g., the link from the power plant site to the water body receiving the cooling system effluent). Although the type of analysis required to decide on whether to use pipes or open channels is a standard one, it does require a careful specification of such items as foundation conditions, excavation costs, local topography, etc.* Because this reflects a level of detail far greater than necessary for our purposes, we abstracted from the issue and postulated that only pipes would be employed.**

The design procedure requires that the following quantities be determined:

- Pipe diameter to carry a specified quantity of flow at a given velocity;
- Pressure drop (commonly referred to as "friction head loss") due to pipe friction;
- Installed cost of pipe of a given diameter and length.

Pipe diameters are computed using the following formula which is a form of the so-called "continuity equation" as used in elementary fluid mechanics [Ref.21,p.86]:

$$d = 12 \sqrt[4]{\frac{L}{449 \pi V}}$$

where

- d = pipe diameter (inches)
- L = flowrate (gpm), and
- V = velocity (ft/sec).

Head loss is computed using the following formula which is based on the Hazen-Williams equation, an empirical relationship widely used in the design of water conveyance systems [6, pp. 37-114]:

$$h = \left[\left(\frac{1.594}{C_{HW}} \right)^{1.85} \frac{D}{d^{4.87}} \right] \left(\frac{L}{449} \right)^{1.85}$$

* For a discussion of the type of analysis required [see Davis and Sorensen: Ref. 6].

** The difference in costs between systems involving pipes and canals is likely to be relatively unimportant, given the necessarily approximate nature of our cost estimates.

where

- h = head loss (feet)
- C_{HW} = Hazen-Williams smoothness coefficient
- D = length of pipe (feet)
- d = diameter of pipe (inches)
- L = flowrate (gpm)

The installed cost per foot of pipe is formed using

$$\alpha_1 d^{\beta_1}$$

where α_1 and β_1 are constants, and d is the pipe diameter in inches. Pitchai [Ref. 17, Chapter 3], using data relevant to municipal water distribution systems, found β_1 to be approximately 1.3. Although we did not attempt to verify this figure, we found it reasonably consistent with values obtained in fitting cost data for water mains presented by Seelye [18]. Using the limited data available in the works of Pitchai and Seelye, and adjusting to 1970 prices, we deduced an α_1 value of approximately 1.2. We recognize that the α_1 value may be higher by as much as a factor of two where poor foundation conditions require extra effort in construction (e.g., pile supports, sheeted excavation, dewatering during construction). We multiply the cost per foot by α_f , a correction factor, to reflect such conditions.

The total installed cost for piping is therefore equal to the cost per foot multiplied by D , the length of pipe, i.e.,

$$\alpha_f \alpha_1 d^{\beta_1} D.$$

A.7.4 Pumps

Required computations relating to pumps include the determination of pump horsepower and capital costs. Concerning the former, it is first necessary to compute h_T , the total head to be overcome by the pump. Depending on the location of the pump and the type of cooling system, this quantity could include pipe friction losses, the elevation head to reach the condenser and the friction losses within the condenser, and the elevation heads associated with the head rejection unit.

Once the head and flowrate (L) are established, the pump horsepower, P , is determined using the following formula based on elementary principles of fluid mechanics [Ref. 18, p.171]:

$$P = \frac{L h_T}{3960 \eta_p}$$

where η_p , the pump efficiency, is an input parameter.

The capital costs for pumps is determined from

$$\alpha_p \beta_p$$

where α_p and β_p are empirical constants. Bauman [Ref. 2] and Guthrie [Ref. 1]] give discussions of the empirical basis for this expression.

A.7.5 Operation and Maintenance Costs

The major items included under operation and maintenance (O & M) are the annual costs attributable to the following: water purchase, water treatment, auxiliary power for cooling system pumps and fans, and maintenance.

The annual cost associated with the purchase of an intake flowrate of L gpm is

$$L \frac{\text{gal}}{\text{min}} \times 8760 \frac{\text{hrs}}{\text{yr}} \times 60 \frac{\text{min}}{\text{hr}} \times \frac{c_w}{10^5} \frac{\$}{\text{gal}}$$

where c_w is the unit cost of water purchase in ¢/1000 gal.*

However, this cost overestimates the annual purchase cost for a number of reasons. One is that steam plants will generally be shut down for some portion of the year to permit routine maintenance. For example, one of the units used for case study purposes will be "down" for 610 hours/year for this purpose. This could be accounted for by multiplying the expression for water purchase cost above by the factor

$$\frac{8760-610}{8760} = 0.93.$$

This correction factor might still lead to an overestimate of the annual water purchase cost for the following reasons:

- (1) Seasonal variations in climate may lead to reduction in the intake flowrate required to maintain the given backpressure while meeting the "design" effluent temperature.

For example, a once-through system might require substantially lower

*While we made no formal estimates for the cost of water intake structures, as a first approximation they could be accounted for by an increase in the unit cost of water purchase.

intake flows in winter, since the temperature of the intake flow will be substantially lower than the value in summer. Recall that L is computed on the basis of a "design" intake temperature which is not meant to be typical of average conditions in either summer or winter.

- (2) Operation of the plant at less than full capacity implies a condenser duty lower than the design value upon which L is based. The lower duty can be dealt with using a lower circulating water flow while still maintaining the given backpressure and meeting the design effluent temperature. One way to adjust the estimate for annual purchase cost is simply to multiply the above expression by a correction factor, F_1 where $0 < F_1 < 1$. After gathering terms, the resulting expression for annual cost is

$$5.25 F_1 L \alpha_w$$

The value of F_1 will be less than or equal to the value resulting from taking account of downtime (e.g., this would be 0.93 for the Hatfield example mentioned above). How much less depends on whether variable speed intake pumps are to be installed, and on the likely impact of seasonal variation in climate and capacity utilization. While it is possible to account for these factors in a systematic quantitative fashion, we chose not to, since the increased computational burden did not appear justified by the likely improvement in the overall cost estimates.*

Similarly, the annual cost associated with the treatment of L gpm is

$$5.25 F_1 L \alpha_{ti}$$

where all symbols are as defined earlier except for α_{ti} which is the unit cost of treatment associated with systems involving heat rejection unit i . (Recall the convention employed regarding the index variable i , namely: $i=1$, mechanical tower; $i=2$, natural tower; $i=3$, cooling pond; $i=4$, no heat rejection unit.) Methods for estimating the unit cost of water treatment have been described by Berg et al. [3], and Lane and Larson [15].

The water treatment cost for helper systems was computed somewhat differently. In such cases, the intake supply received the same treatment as a once-through system (i.e., $i=4$), and added to this was the cost to provide additional treatment for that portion of the intake that passed through a cooling tower or pond before

*For a description of a systematic accounting of these factors see Fuller [Ref.11].

discharge. This latter cost is computed as

$$K_{hi} \ 5.25 \ F_1 \ L' \ a_{ci}$$

where L' is the flowrate (gpm) passing through the tower or pond, a_{ci} is the unit cost, and K_{hi} is an input parameter that is less than or equal to unity to allow for the fact that the unit cost of treatment associated with heat rejector i may be less than a_{ci} given that the outlet flow is not to be recirculated.*

The costs associated with auxiliary power for cooling system pumps and fans is computed in two parts. First, there is the actual cost incurred to supply energy to run the pumps and fans.** This is computed as

$$0.746(P) (F_1) (8760) (\gamma_1),$$

where P is the combined horsepower of all pumps and fans, γ_1 is the penalty cost for energy (\$/kwhr), and F_1 is as defined above. The conversion factors represent the number of kilowatts per horsepower (0.746) and the number of hours per year (8760).

Second, there is the cost associated with the fact that plant capacity is decreased is the auxiliary power requirements, as given by P , are greater than those in the cooling system associated with the privately optimal design (i.e., P_p). This is computed as

$$0.746 (P - P_p) \gamma_2$$

where γ_2 is the penalty cost for capability loss in \$/kw-yr.

While the nature of the penalty costs may be different for different utilization, it is not unreasonable to expect that such coefficients will be obtainable. In fact, traditional engineering studies for power plant design generally contain reference to penalty costs for both energy and capacity, i.e., the values of α_1 and α_2 , respectively, required as input for our computer program.

The annual maintenance cost (i.e., labor and routine equipment replacement) is taken as a fraction of the total capital cost for all components of the cooling system; this fraction is given as α_{mi} (an input parameter), where the index i refers to the heat rejection unit type.

The total O & M cost is taken as the sum of the components described above.

* We set $K_{h3} = 0$, since flow passing through ponds in a helper system would not generally receive any treatment beyond that used in once-through systems.

** We abstracted from that fact that, for mechanical draft towers, the four may not have to run at a constant level while the unit is operating. This abstraction seems reasonable given the level of detail employed in other parts of the analysis.

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APPENDIX B

DERIVATION OF COOLING TOWER CAPITAL COST ESTIMATING FUNCTIONS

Over the years a number of investigators have presented graphical techniques for estimating the cost of cooling towers [1, 3]. While such techniques could have been employed directly, we found it far more convenient for computational purposes to reduce one such graphical method to a more efficient procedure involving algebraic equations. This appendix describes the particular cost estimating procedure we adopted, and the results of our efforts to convert the particular graphs involved into algebraic formulas.

Mechanical Draft Towers

We adopted the approach of Dickey and Cates who presented a cost estimating procedure for a crossflow mechanical draft cooling tower typical of those used for large steam-electric power plants [2]. Their approach is based on the computation of a quantity called the "required tower units" (TU), which can be interpreted as the "synthetic square feet of heat transfer hardware"; i.e., a quantity analogous to square feet of condenser surface area.

Dickey and Cates presented data from nine current power plant projects indicating that the cost of mechanical towers per tower unit is fairly uniform. The average value, in dollars per tower unit, was 4.97, and the standard deviation was 0.57. Consequently, to estimate the capital cost of a proposed mechanical draft tower, we first computed the required number of tower units, and then multiplied it by 4.97. Below we describe the procedure used in computing the required tower units.

Following Dickey and Cates, the required tower units, TU, is defined by the following equation:

$$TU = L \cdot RF,$$

where L is the flowrate in gallons per minute, and RF is the "rating factor." The latter is a measure of the "relative degree of difficulty" in cooling one gallon per minute of flow. For example, an $RF = 1.1$ is interpreted as corresponding to a system requiring 10 percent more fill plan area than a system with an $RF = 1.0$.

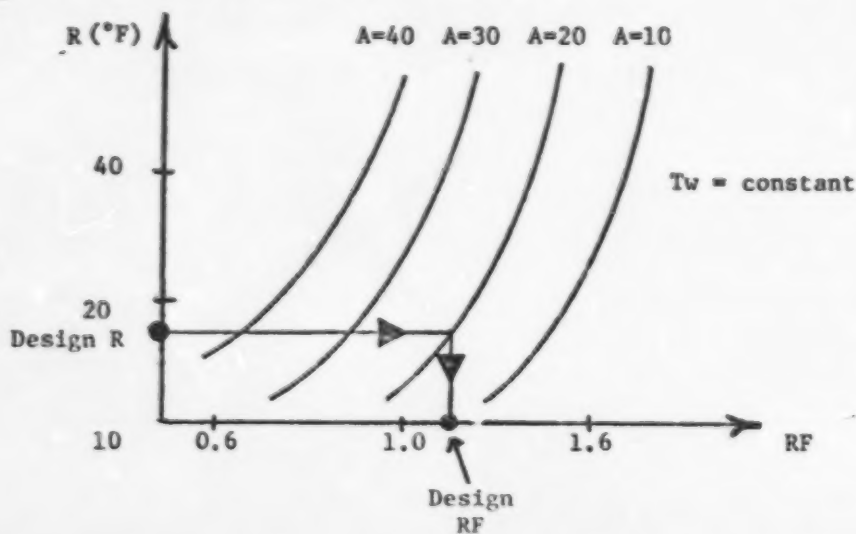
The rating factor may be determined graphically from a series of curves that represent the outputs from an analysis of the heat transfer characteristics of various tower designs. All that is required to determine RF is the specification of the design values of the following:

R = range (i.e., inlet temperature minus outlet temperature)

A = approach (i.e., outlet temperature minus wet bulb temperature)

T_w = wet bulb temperature.

The figure below indicates the general shape of the curves for a given T_w used to determine RF . (Dickey and Cates presented 10 such figures corresponding to different values of T_w .) RF is thus determinable for any given T_w and any allowable combination of alternative design values of Range and Approach, as illustrated in the figure for $R=17$ and $A=20$.



For purposes of using a computerized cooling system design procedure, there are two shortcomings associated with this graphical approach to determining RF. The first is that it would require the input data for the cooling system design computer program to contain an ordered list of rating factors corresponding to a large combination of range, approach and wet bulb temperature to be examined in the design procedure. The second is that a multivariable interpolation procedure would be required to obtain rating factors for values of the design variables not explicitly represented by the Dickey-Cates figures.

To overcome these shortcomings we performed a statistical regression analysis to develop estimating equations which approximated the curves presented by Dickey and Cates. To accomplish this we first constructed a table of what statisticians would refer to as raw data. In particular, we selected some 400 points from the Dickey-Cates figures and recorded the associated values of RF, A, R, and Tw.

Before we could continue with the statistical analysis of this data, it was necessary to specify a form for the estimating equation. Experience in engineering cost estimating for individual pieces of hardware suggested that a power function might be appropriate. Hence, we first tried an equation of the form

$$RF = a_0 R^{a_1} A^{a_2} Tw^{a_3} ,$$

where a_0 , a_1 , a_2 and a_3 are parameters to be estimated in the regression analysis. The logarithm of this function is linear in the parameters and consequently we could rely on the large body of theoretical and computational methodology available for performing linear regression analysis by the method of "least-squares."

The "least squares" analysis yielded the following estimating equation:

$$RF = \frac{e^{5.837} R^{0.652}}{A^{0.997} Tw^{1.237}}$$

where $e = 2.718$. The square of the multiple correlation coefficient is 93.646 percent.

Although the multiple correlation coefficient associated with the power function is fairly high, the shape of the curves in the Dickey-Cates paper suggested that a second order polynomial might also serve well as a functional form. A least squares analysis using a second order polynomial was performed, and the square of the multiple correlation coefficient turned out to be 98.246 percent, a significant improvement over the value associated with the power function. The regression equation, which is the one employed in the cooling system design computations discussed herein, took the following form:

$$\begin{aligned} RF = & 3.24 + 0.000898 \text{ TwA} + 0.137R - 0.000169 \text{ Tw}^2 \\ & - 0.000707 R^2 - 0.0451 \text{ Tw} - 0.147A \\ & + 0.00164 A^2 - 0.000817 \text{ RTw} - 0.00135 \text{ RA.} \end{aligned}$$

Hyperbolic Natural Draft Towers

For hyperbolic natural draft towers, as in the case of mechanical draft towers discussed above, we adopted an approach due to Dickey and Cates [2]. They presented a graphical technique for estimating the cost of hyperbolic natural draft towers typical of those used in large steam-electric power plants constructed in the late 1960's.

The procedure used by Dickey and Cates is based on curves relating the unit cost of a tower to the range and approach. They presented twelve such curves corresponding to different values of relative humidity and range.

To estimate the unit capital cost, C , associated with a natural draft tower ($\$/10^3 \text{ Btu/hr.}$), it is necessary to specify the design values of the following parameters: R , A , Tw (defined above) and R_h , the relative humidity given as a percent. Whereas the relative humidity does not play a role in estimating

the capital costs for mechanical draft towers, it does have a substantial impact on the art of natural draft towers.

Once the design values of R , A , T_w and R_h are specified, it is possible to use the figures of Dickey and Cates (interpolating where necessary) to deduce the unit capital cost in $\$/10^3$ Btu/hr. To find the total capital cost, it is only necessary to multiply the unit cost by the number of 10^3 Btu/hr. to be transferred by the tower.

For reasons of computational convenience, as outlined above in the discussion of mechanical draft towers, we decided to convert the procedure of Dickey and Cates to one involving equations instead of graphs. Following much the same approach as in the mechanical tower case, we abstracted some 600 points from the figures given by Dickey and Cates. Each point corresponded to a set of values for C , A , R , T_w and R_h .

The first step in fitting these data involved the use of an estimating equation of the form

$$C = b_0 R^{b_1} A^{b_2} T_w^{b_3} R_h^{b_4},$$

where $b_0, b_1, \dots, b_4$ are parameters to be estimated. The least-squares regression analysis to determine the parameter values yielded the following equation:

$$C = \frac{e^{14.6}}{R^{0.756} A^{1.307} T_w^{1.650} R_h^{0.285}}$$

The associated square of the multiple correlation coefficient is 98.89 percent.

As in the case of mechanical draft towers, we also fit the data points with a second order polynomial. The result from this analysis was:

$$\begin{aligned} C = & 0.971 + 0.00264 RA + 0.00237 T_w A \\ & + 0.000238 R R_h - 0.180R - 0.0462 R_h \\ & - 0.621A + 0.00569 A^2 + 0.000853 R_h A \end{aligned}$$

$$\begin{aligned}
 & - 0.00321 T_w^2 + 0.371 T_w + 0.000889 R^2 \\
 & + 0.000126 R_h^2 + 0.000314 R T_w - 0.000105 R_h T_w.
 \end{aligned}$$

However, the square of the multiple correlation coefficient for the polynomial fit is 97.02, a value lower than the corresponding value for the power function. Consequently, we used the power function in estimating the cost of hyperbolic natural draft towers.

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APPENDIX C

DESIGN DATA FOR 547 MW COAL-FIRED UNIT CASE STUDY

C.1 Input Values for Cooling System Design Analysis

Following are listed values for important input parameters for the cooling system design analysis that were held constant for all 12 runs of the cooling system design program as discussed in Section II. Input parameters for cooling system design that changed with each steam cycle case run were previously listed in Table 2 of Section II. Unless otherwise indicated in the right-hand column, the values utilized as input were either the same values utilized as design conditions for the prototype unit or else values derived from the component selection results of the prototype. Bracketed [] reference numbers refer to references listed under A.8 of Appendix A. For further discussions on variable definitions and uses, see Appendix A.

<u>Variable Name and Description</u>		<u>Values Employed and Comments</u>	
[A. <u>Site Specific Climatologic and Hydrologic Conditions</u>]			
BARL	Maximum design intake flowrate (gpm)	500,000	Arbitrarily selected by CEM; approx. 2x design intake of prototype
TA	Design ambient temperature of intake water (°F)	74°	Prototype summer design value
TWB	Design wet bulb temperature (°F)	73°	95% annual exceedance value [The Marley Co.]
TDEW	Design dew point temp. (°F)	60°	Conservative values based on July conditions [Ref. 20]
RH	Design rel. humidity (%)	50	
SORAD	Design daily solar radiation (Btu/ft ² /day)	1820	
WSPEED	Design windspeed (miles/hr)	8	
[B. <u>Condenser Components and Costing Factors</u>]			
CFUDG	Empirical constant in heat transfer derivation	550	[Ref. 1]
VEL	Velocity of cooling water	5.0	
ACON	Coefficients in condenser capital cost function	1.2	Vendor price lists
BCON		75,000	
CCON		0.3	

Variable Name and Description		Values Employed and Comments	
[B. <u>Condenser Components and Costing Factors - Continued</u>]			
BCON	Inside diameter of condenser tubes (inches)	.777	
DOCON	Outside diameter of condenser tubes (inches)	.875	
TUBEL	Length of a single condenser tube (ft.)	55	
FPASS	Number of passes in condensers	1	
RISPMO	Minimum temperature increase through condenser (°F)	3	Assumed
[C. <u>Heat Rejection Component and Costing Factors</u>]			
TUDOL	Unit cost of Mechanical Draft tower. \$ per "tower unit"	}	See Appendix B
AM(I)	Coefficients in function for M.D. tower rating factor		
CN,RN, AN,WN, EN	Coefficients in Natural Draft tower capital cost function		See Appendix B
CTLD	Unit cost for land, site preparation and construction of dikes or dam (\$/surface acre)	\$5,000	Assumed
RISMM	Minimum temperature decrease (range) across tower (or pond)	10	} Constrained by available data
APMM	Minimum "approach" to wet bulb (°F)	8	
[D. <u>Pipes and Pumps</u>]			
PLEN(I) I=1-9	Pipe lengths--keyed to system layout diagrams (ft)	Various: 25-650	Prototype and assumed
CHW	Hazen-Williams pipe smoothness coefficient	130	
APIPE	Multiplicative constant in pipe capital cost function	1.2	} [Ref. 17]
BPIPE	Exponential constant in pipe capital cost function	1.3	
PFUDG	Special constant to account for variable foundation conditions	1.0	

<u>Variable Name and Description</u>		<u>Values Employed and Comments</u>	
[D. Pipes and Pumps - Continued]			
HEAD(I) I=1-4	Elevation heads--keyed to system types (ft)	Various: 0-35 }	Prototype and assumed values
APUMP	Multiplicative constant in pump capital cost function	1,560	
BPUMP	Exponential constant in pump capital cost function	0.7	
PEFC	Pump design efficiency (Z/100)	0.8	
[E. Water Supply and Treatment - Related Factors]			
CTWTS	Unit cost of purchased water (¢/1000 gal.)	0.0¢	
TRTCT(I)	Unit cost of water treatment, using heat rejector i (¢/1000 gal)	3.0¢	[Ref. 15]
WIND(I)	Wind loss as a fraction of flow through heat rejector i (i,i=1,...,3)	.001 or 0.0 }	Assumed
XNC(I)	Number of cycles of concen- tration in recirculating cooling system	5	[Refs. 4;7]
[F. Other Cost Related Parameters]			
XLFF	O&M cost correction factor	0.915	
XFCR	Annual fixed charge rate (ap- plicable to all capital cost items)	0.114	
PCTPF	Penalty cost for energy for pumps and fans (\$/kwhr)	0.0023	
YRPEN	Penalty cost for capability loss (\$/kw/yr)	\$12.5	Based on \$110/Kw x .114
CTMT(I)	Annual maintenance cost factor-- as a fraction of total capital cost for system involving heat rejector (i,i=1,...,4)	.008 to .015	

C.2 Additional Detail on Steam Cycle Design Modifications

Case 1 - Base Plant

This is the base-case described above. It has a single reheat (steam conditions of 3500 psig, 1000F/1000F) and a dual pressure condenser (back pressures of 1.72 and 2.14" Hg abs.).

Case 2 - Double Reheat - 1.72"/2.14" Vacuum

This case is considered the most efficient practical design with a 1.72"/2.14" vacuum condition. It is a double reheat cycle (steam conditions of 3500 psig, 1000F/1000F/1000F) with seven feedwater heaters, pump forward drains and a separate turbine drive for the boiler feed pump. The major modification from Case 1 is the provision for the double reheat which includes the boiler, turbine and additional reheat piping. There is a major capital cost addition for both the boiler and the turbine.

Case 3 - Single Reheat - 1.72" Vacuum

This case is a modification of the base plant using a single pressure condenser at 1.72" vacuum instead of a 1.72" and 2.14" dual pressure design. Feedwater heaters are modified and adjusted for the colder condensate and change in water rate, but all other components remain the same as in Case 1.

Case 4 - Double Reheat @ 1.72" Vacuum

This case is considered the most efficient practical design with a 1.72" single vacuum condition. With the exception of the vacuum, the changes from the base plant are similar to those described in Cases 2 and 3 above.

Case 5 - Single Reheat - 2.50"/3.05" Vacuum

The fifth case involves a four-flow, tandem-compound turbine with a smaller exhaust end for the 2.50"/3.05" dual pressure condenser than the base plant's turbine. This results in a substantially lower turbine cost for Case 5 as compared to Case 1. The feedwater heater surface is lower than that in Case 1 to reflect the hotter condensate; however, the other heaters are increased by a small percentage to handle the increased water rate. With the exception of the turbine, feedwater heaters, and condensers, the unit is the same as the base plant Case 1.

Case 6 - Double Reheat - 2.50"/3.05" Vacuum

This case is considered the most efficient cycle practical using a 2.50"/3.05" vacuum. The turbine has the same exhaust end as Case 5. The cycle is a double reheat 3500 psig, 1000F/1000F/1000F with seven feedwater heaters, pump forward drains, and a separate turbine drive for the boiler feed pump. The major modification

to Case 5 is the provision for double reheat which influences the boiler, turbine, and additional reheat piping.

Case 7 - Single Reheat - 2.50" Vacuum

This case is a modification of Case 5 using a single pressure condenser at 2.50" vacuum. With the exception of feedwater heaters which are adjusted to account for changes in the temperature of the condensate and water rate, this design is the same as Case 5.

Case 8 - Double Reheat - 2.50" Vacuum

This case is considered the most efficient practical design with a 2.50" vacuum. With the exception of the vacuum, the changes from Case 5 are similar to those described in Case 6 above.

Case 9 - Single Reheat - 3.25"/3.95" Vacuum

The major difference between this case and the base plant, Case 1, is in the turbine which is modified as a result of the difference in condenser vacuums. Case 9 employs a four flow tandem compound turbine with a smaller exhaust end than the turbine in the base plant. The feedwater heater surface is lower than that used in Case 1 to reflect the hotter condensate; however, the other heaters are increased by a small percentage to handle the increased water rate. With the exception of the turbine, feedwater heaters, and condensers, the design is the same as Case 1.

Case 10 - Double Reheat - 3.25"/3.95" Vacuum

This case is considered the most efficient cycle presented using a 3.25"/3.95" vacuum. The turbine has the same exhaust end as Case 3. The cycle is double reheat 3500 psig, 1000F/1000F/1000F. The major modification to Case 9 is the double reheat which influences the boiler, turbine and additional reheat piping. There is a reduction in the size of all heaters (as compared to Case 9) owing to the reduction in water rate and slight increase in the superheat of the extraction steam.

Case 11 - Single Reheat - 3.25" Vacuum

This case is a modification of Case 9 using a single pressure condenser at 3.25" vacuum. With the exception of the feedwater heaters which are adjusted to account for changes in the temperature of the condensate and water rate, this design is the same as Case 9.

Case 12 - Double Reheat - 3.25" Vacuum

This case is considered the most efficient practical design with a 3.25" vacuum. With the exception of the vacuum, the changes from Case 9 are similar to those described in Case 10 above.

APPENDIX D
DESIGN DATA FOR 1000 MW NUCLEAR UNIT CASE STUDY

Following are listed values for important input parameters for the cooling system design analysis that were held constant for all 3 runs of the cooling system design program as discussed in Section III. Input parameters for cooling system design that changed with each steam cycle case run were previously listed in Tables 12 and 13 of Section III. Unless otherwise indicated in the right-hand column, the values utilized as input were either the same values utilized as design conditions for the prototype unit or else values derived from the component selection results of the prototype. Bracketed [] reference numbers refer to references listed under A.8 of Appendix A. For further discussions on variable definitions and uses, see Appendix A.

<u>Variable Name and Description</u>		<u>Values Employed and Comments</u>	
[A. Site Specific Climatologic and Hydrologic Conditions]			
BARL	Maximum design intake flow-rate (gpm)	1.7x10 ⁶	Arbitrarily selected by CEM; approx. 2x design intake of prototype
TA	Design ambient temperature of intake water (°F)	75°	Prototype summer design value
TWB	Design wet bulb temperature (°F)	75°	95% annual exceedance value [The Marley Co.]
TDEW	Design dew point temp. (°F)	62°	Conservative values based on July conditions [Ref. 20]
RH	Design relative humidity (%)	55	
SORAD	Design daily solar radiation (Btu/ft ² /day)	1700	
WSPEED	Design windspeed (miles/hr)	9	
[B. Condenser Components and Costing Factors]			
CFUDG	Empirical constant in heat transfer derivation	450	[Ref. 1]
VEL	Velocity of cooling water (ft/sec)	6.0	
ACON	Coefficients in condenser capital cost function	1.2	Vendor price lists
BCON		75,000	
CCON		0.54	
DCON	Inside diameter of condenser tubes (inches)	.902	

<u>Variable Name and Description</u>		<u>Values Employed and Comments</u>	
[B. <u>Condenser Components and Costing Factors - Continued</u>]			
DCON	Outside diameter of condenser tubes (inches)	1.0	
TUBEL	Length of a single condenser tube (ft.)	50	
FPASS	Number of passes in condensers	1	
RISMNO	Minimum temperature increase through condenser (°F)	3	Assumed
[C. <u>Heat Rejector Component and Costing Factors</u>]			
TUDOL	Unit cost of Mechanical Draft tower. \$ per "tower unit"		See Appendix B
AM(I)	Coefficients in function for M.D. tower rating factor		
CN,RN, AN,WN, HN	Coefficients in Natural Draft tower capital cost function		See Appendix B
CTLD	Unit cost for land, site preparation and construction of dikes or dam (\$/surface acre)	\$5,000	Assumed
RISMM	Minimum temperature decrease (range) across tower (or pond)	10	Constrained by available data
APMM	Minimum "approach" to wet bulb (°F)	8	
[D. <u>Pipes and Pumps</u>]			
PLEN(I) I=1-9	Pipe lengths--keyed to system layout diagrams (ft)	Various: 25-75	Prototype and assumed
CHW	Hazen-Williams pipe smoothness coefficient	130	
APIPE	Multiplicative constant in pipe capital cost function	1.2	[Ref. 17]
BPIPE	Exponential constant in pipe capital cost function	1.3	
PFUDG	Special constant to account for variable foundation conditions	1.0	

<u>Variable Name and Description</u>		<u>Values Employed and Comments</u>	
[D. Pipes and Pumps - Continued]			
HEAD(I) I=1-4	Elevation heads--keyed to system types (ft)	Various: 0-35	} Prototype and assumed values
APUMP	Multiplicative constant in pump capital cost function	1,560	
BPUMP	Exponential constant in pump capital cost function	0.7	
PEFC	Pump design efficiency (%/100)	0.8	
[E. Water Supply and Treatment - Related Factors]			
CTWTS	Unit cost of purchased water (¢/1000 gal.)	0.0¢	
TRTCT(I)	Unit cost of water treatment, using heat rejector i (¢/1000 gal)	0.01¢ to 1.8	} [Ref. 15]
WIND(I)	Wind loss as a fraction of flow through heat rejector i (i,i=1,...,3)	.001 or 0.0	
XNC(I)	Number of cycles of concen- tration in recirculating cooling system	5	[Refs. 4;7]
[F. Other Cost Related Parameters]			
XLF	O&M cost correction factor	0.90	
XFCR	Annual fixed charge rate (ap- plicable to all capital cost items)	0.158	
PCTPF	Penalty cost for energy for pumps and fans (\$/kwhr)	0.0020	
YRPEN	Penalty cost for capability loss (\$/kw/yr)	\$30.0	Based on \$110/Kw x .114
CTMT(I)	Annual maintenance cost factor-- as a fraction of total capital cost for system involving heat rejector (i,i=1,...,4)	.002 to .012	